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STUDY OF ARMY BUILDING VENTILATION SYSTEMS FOR ENERGY CONSERVAT--ETC(U)

AUG 78 J R WROBEL DAAK70-77-C-0217

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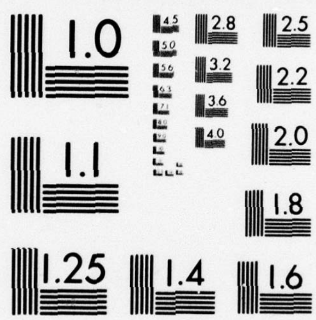

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STUDY OF ARMY BUILDING VENTILATION SYSTEMS FOR ENERGY CONSERVATION

Jacobs and Associates, Inc.
28 Research Drive
Hampton, A 23666

1 August 1978

Final Report

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Prepared for:
Research and Technology Division
USA Facilities Engineering Support Agency
Fort Belvoir, VA 22060

Tenant Division
USA Mobility Equipment R&D Command
Fort Belvoir, VA 22060

ERRATA SHEET FOR FESA-RT-2055

1. Page 9 - "The referenced guidelines describe the priorities for air conditioning based on the average annual hours that specified dry and wet bulb temperature limits are exceeded".

Change to read: "Department of Defense guidelines describe the priorities for air conditioning based on the hours that specified dry and wet bulb temperature limits are exceeded."

2. Page 18 - "Engineering Weather Data, Air Force, Army and Navy Manual for Facility Design and Construction (1967)".

Change to read: "Engineering Weather Data, Air Force, Army and Navy Manual for Facility Design and Planning (1978)".

3. Page 33 - after "(SCFM)" add: "(Standard Cubic Feet per minute)".

4. Page 35 - after "MBTUY" add: "(Million British Thermal Units per year)".

5. Page 39 - "The energy lost due to a loss of conditioned air in the faction (f) of the air handling capacity is given as follows from the prior analysis on vent loss.

E (annual BTU/CFM airflow) $= ft \; W_H \times 1.08 (\bar{T}i - \bar{T}o) + Wc \times 4.5 \, Hc)$
where f is the fraction of leakage, it is the hours per week that the air handler system operates in the leakage condition. The other elements of the function are provided for typical cities in the CONUS in Table (5) charts."

Change to - "The energy lost due to a loss of conditioned air in the fraction (f) of the air handling capacity is given as follows:

E (annual BTU/CFM airflow) $- ft \; Wh \times 1.08 (\bar{T}i - \bar{T}o) + Wc \times 4.5 \, Hc$
where:

f = the fraction of leakage
t = hours per week air handler operates in leakage condition
Wh = average heating season in weeks per year
Ti = average indoor temperature
To = heating season average outdoor temperature
Wc = average cooling season in weeks per year
Hc = average cooling season enthalpy difference (BTU/Lb).

Typical values for these parameters are given for some cities in CONUS in Table (5).

6. Page 42 - after the sentence "Therefore the winter loss for a full time system is for a 26 week heating period."

Add: "$E\ \frac{\text{Annual BTU}}{\text{CFM}} = (26\ \frac{\text{week}}{\text{Yr}})\ (L)\ (168\ \frac{\text{Hr}}{\text{Week}})\ (1.08\ \frac{\text{BTU}}{\text{CFM-HR-°F}})(20°F + \frac{20°F}{2.0})$

$= 141{,}500\ \frac{\text{BTU}}{\text{Yr}} \times L$

or "

7. Page 44 - "In hydraulic systems the circulated _ _ _ _ ".

Change to: "In hydronic systems the circulated _ _ _ _".

8. Page 63 - " _ _ _ the ducts will reflect a raise or drop in static pressure _ _ _ ".

Change to: "_ _ _ the ducts will reflect a rise or drop in static pressure.

9. Page 63 - after "_ _ _ _ cooling season hot deck reset °F and ΔHc is the cooling season cold deck enthalph reset."

ADD: "As before, W_H is the heating weeks and Wc is the cooling weeks."

10. Page 65 - Change "The energy savings is computed as follows for a typical installation:"

To: "The energy savings can be estimated by multiplying the % of off time by the KW rating of the compressor and the cooling hours, or as follows:"

11. Page 77 In the sentence: "This does not allow reduction below the recommended minimum values for the application", delete the word "recommended".

12. Page 83 - In the sentences: "Prone and at rest the oxygen demand is about .01. At heavy work the rate is about .05."

Add: "SCFM" after ".01" and ".05"

13. Page 84 - In the sentence: "Therefore the basic physiological need for long or steady occupancy is from 1.6 to 8CFM with 5 CFM as an acceptable safe criterion average"

After 8 CFM, add "of outdoor air"

14. Page 86 - In the sentence: "Typical average face velocity thru these filters is in the range of 250 FPM to reduce velocity and to reduce pressure drop."

Delete: "to reduce velocity and to reduce pressure drop"

15. Page 133 - In the sentence: "Another aspect to be considered in R&D goals is that the government buildings ----."

after the words "R&D goals" add: "of the HVAC industry"

16. Page 134 - In the sentence: " ---- and also design to energy effectiveness through the use of existing programs for optimal pipe insulation ----."

After the word "existing" add: "energy conservation"

17. Page 135 - After the sentence: "Single building and small facility sites are not currently conducive to cost effective application of EMC equipment of the complexity that is prevelant."

Add: "This is being studied by several Corps of Engineers District Offices for the Office, Chief of Engineers."

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A generalized analysis of fixed base facility ventilating equipment is presented to describe practical technical and cost-effective energy conservation goals and prospects for future equipment development.

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OBJECTIVE OF THE STUDY

The objective of this technical study was to evaluate and recommend options to reduce energy losses due to Army building heating, ventilating, and air conditioning (HVAC) systems ineffectiveness. The options considered were comparatively evaluated within prudent constraints of technical feasibility and economic justification for applicability to both new and rehabilitated facility projects in the future. The spectrum of Army fixed-base building mission functions included: barracks, training facilities, administration buildings, recreational buildings and storage buildings. Since the objective is to provide guidance and direction recommendations in general for buildings of varying age, architecture, use and geographical location, it was determined that the study direction should be to provide objectives for application of technology extensions and research and development goals or targets. Suggestions for innovative changes in HVAC controls, equipment, filters and occupancy criteria achievable in the near term with rapid payback prospects and concurrent energy savings to justify the recommendation were to be sought. The contemporary guidelines for the establishment of economic justifications were those extracted from pertinent DOD military construction guidance, including items such as the Energy Conservation Investment Program (ECIP) Guidance from the office of the (Deputy) Assistant Secretary for Defense for Installations and Housing.

To aid in the focus of the application of resources to the study, several assumed constraints were incorporated to accomplish the objective within the allotted level of effort, time, and generalized

nature of the potential use of study recommendations. These limit constraints are outlined here to eliminate concerns that the study effort did not adequately address items of contemporary interest to the Government in military and civil works project energy conservation efforts.

Architectural features of buildings which could influence energy effectiveness were not considered. The principal reason was that these features are generally not the prerogative of the HVAC engineer or contractor to alter. Also, such suggestions or recommendations must always be specific to a given building, site, and geographical (climatic) condition. Therefore, our discussion does not encompass items such as double-entry weather locks, solar screens, multi-glazing, wall insulation, or other numerous features that have economic merit regarding energy cost avoidance and capitalization payback. It is assumed that these non-HVAC prospects are to be pursued by the appropriate discipline group. It is intended that this study outcome could be used to assist system level decision managers to weigh the relative merits of both the HVAC system prospects for energy savings and those of the architectural system potentials to reach peak cost-effectiveness on the overall facility utility investment program.

Another technical area which we did not address was the influence of the primary converter effectiveness or choice of fuel. The guidance of the DOD on the future limitation of electric heat, natural gas use, and small A/C units for building cooling were incorporated. Recommendations to convert from oil to coal or other similar primary source type of energy were not appropriate for this study. The choice of fuel and the efficiency of its use up to the entry into the specific building HVAC system is dependent upon many parameters not possible to cover in any

general study. For example, consider the following situations: A small building may be served by a small unit heater of relatively low conversion efficiency while the same building, at another base, could be served by district steam from a large and efficient converter. Therefore, the recommended HVAC system actions cannot assume that the cost of changing the fuel source converter for a small building is necessarily in proportion to the individual building needs. Also, the relative availability and local market conditions affect energy costs at the input point in different regions of the CONUS. Therefore, the study cannot categorically recommend coal, oil, or gas source for any given size of building or application independent of the site specifics. The conversion efficiency from the fossil fuel source is not a specific influence on HVAC options at the level of the interest of this study, since, once converted, the energy losses in the buildings relate to HVAC system ineffectiveness and misallocations.

Certain HVAC system equipment was also excluded from analysis in the study. These included heat pump systems and also solar source systems. These systems are currently under analysis by numerous groups to define applications and R&D goals. Both were avoided in this study because of similar and related reasons. First, the locality or region of application is quite specific to these systems. Both systems mentioned interact with the environment and are, therefore, specific to a location or site. Conclusions drawn as to effectiveness at one site do not apply at another site for a similar building type and size. Also, these systems are intimately connected with building construction details since solar and heat pump systems are power limited due to installed capacity limits and exterior ambient

conditions. These systems rely upon thermal inertia, either that associated with the building mass or due to a thermal sink installed to match input with output. Also, most of both types of applications use a redundant source to make up the energy budget under extreme operating conditions outdoors. The heat pump and solar absorbers have also been combined in numerous ways to effect energy savings. The options available are quite numerous and each is worthy of exhaustive application analysis. Often the HVAC system design resulting from these combinations includes the requirement to select specific efficiency equipment or can only be cost competitive under some arbitrary set of future constraints or energy source, availability and cost. In many cases, the heat pump and the solar system rely upon electrical resistance heat as a backup. This is perhaps adequate for isolated applications within a utility network. However, if the overall utility peak load to average load ratio is high, then a cost premium will be applied to the use of peak-use power to absorb the capital equipment costs of peak demand matching and large periods of operation at reduced output. In many cases, utilities install low-efficiency, premium fuel rolling reserves such as natural gas fueled gas turbines. In these cases, the peak demand use will draw upon inefficiently used and scarce fuels to augment the basically efficient but variable output heat pump and solar system equipment. Within the scope of this study it was not practical to examine these myriad options for heat pump and solar systems. These options are currently being addressed in the general trade and professional literature at considerable length.

SUMMARY

In this report, the technical analysis and evaluation performed over the active life of the contract is presented in summary form to provide guidance and information on potential Army building HVAC energy use improvements.

The report begins with an evaluation of the scope of the Army building systems that prevail and also those that are outside the bounds of interest to this general study. The first half of the report deals with cost and performance evaluations of systems and equipment with emphasis on generally applicable improvements and trade-offs for most applications of interest. In particular, the effect of the control on outside air in so called "economizer" cycles is discussed in depth to illustrate the potential for savings in energy for various CONUS locations. Methods are provided to analyze the energy losses and/or savings practical to achieve with the improved equipment and controls that are suggested.

The second half of the report deals with the use and control of outside air for ventilation purposes. The physiological and safety requirements for human occupancy are discussed. The conversion of these to volume flow rate ranges for fresh air supply are presented for conditions with and without effective particle filtration. The various heat recovery cycles for make-up ventilation air are analyzed and discussed with examples to illustrate the energy saving potentials. The types and operation of filters to improve recirculated air quality are compared along with costs of typical equipment per unit capacity. The final section deals with the potential for R&D improvements in

HVAC equipment and methods of control to achieve better energy utilization in Army buildings.

SURVEY AND DEFINITION OF ARMY BUILDING HVAC SYSTEMS

The purpose of this task was to catalog and evaluate the variety of system design features prevalent in Army building HVAC equipment at this time. This type of evaluation cannot be specific to each individual system because there are very nearly as many system variants as there are buildings, which number in excess of 100,000. The goal was to enumerate the types of generic systems with the objective to address common features insofar as possible so that facility engineers could identify prospects for improvement among their plant responsibilities. The mutual elements prevalent in most systems include plumbing, ducting, related insulation, air movers, fluid movers, heat exchangers, and controls. This task permitted the allocation of needed resources to the systematic review of the fundamentals of the design and the basis for design for the many and varied system types which have been installed over the years in military fixed base facilities. It is recognized that many buildings were constructed for a given mission function which has been changed perhaps more than once over the active life of the building and the current or future mission may not be that of the initial use. Also, many buildings are already being used beyond their planned life span and renovations, particularly of the HVAC systems, are essential to achieve operational viability in light of energy costs.
There have been DOD guidelines issued to prescribe the limits on the use of scarce and/or expensive energy sources in specific MILCON applications. These guidelines are subject to periodic revision and update as new data becomes available and, therefore, specific design criteria are not repeated here. Suffice it to say that: electric resistance heat is

discouraged at all levels of size scale; reheat cycles using primary energy are discouraged; small, independent, inefficient package air conditioners are use limited; natural gas use is curtailed; and coal firing for large district energy applications is encouraged.

The above noted system elements are typical of the design of almost all HVAC systems although not all systems include all features. The energy utilization of the HVAC system can be improved by selecting the equipment that is to be replaced or modified within a given system in rehabilitation projects. For example, it may be economically justified to reinsulate duct and pipe for a system and to improve controls, but not to replace the duct itself or the pipe based on the economic and planned life of the building.

The generic types of HVAC systems that have been used in DOD construction and similar institutiónal applications are listed in Table 1 according to common factors. The first major categorization is central systems versus unitary or packaged systems. In the central systems there are all air, all fluid, and combination system. The referenced Table 1 provides self-explanatory detail as to the types of systems. It should be noted that the table contains options with "reheat" as a major or minor factor in the operation. It is recognized that new and retrofit construction will eliminate the majority of these options due to their intrinsic inefficiency of energy utiIization. The nomenclature used in the Table 1 is consistent with industry conventions, and should be self-explanatory.

Each of the HVAC systems represented in Table 1 have application in a specific class of building size and function. In the array of Army

building types, by use, the environmental requirements influence the selection of an HVAC system that meets the criteria. Recent DOD directives have revised environmental requirements and provide for non-use period set backs on the HVAC equipment for temperature and ventilation. The climatic zone in which a building is located will influence the choice as well. The referenced guidelines describe the priorities for air conditioning based on the average annual hours that specified dry and wet bulb temperature limits are exceeded. The annual average mean degree days, as measured from a 65^{o} F base, has a similar effect on the heating system selected for a given application, since the annual operating hours (average) and the installed capacity are strong drivers in the life cycle cost analysis and the cost/benefit analysis of conservation measures.

Although a large array of system configurations is presented in Table 1 there are common elements in each which can be evaluated separately and the composite system cost and function built up as an accumulation of the parts. The common elements of major significance are:

fluid piping and valves
piping insulation
air duct
air duct insulation
fluid pumps, including compressors
air fans, air handlers
heat exchangers, converters, boilers
controls and related operators

TABLE 1. TYPICAL ARMY BUILDING HVAC SYSTEMS

Central Systems

All Air System	Air with Water/Steam	All Water/Steam
Single Path	Water	Water
Single Duct, Constant Volume	Terminal Fan Coil Unit	Fan Coils, Baseboard
		Cabinet
High Velocity	Two pipe changeover	exhaust & infiltration
Low velocity	Three pipe	wall apertures
	Four pipe	ventilation ducts
Single Duct, Variable Volume	Baseboard, Cabinet, Fin Tube	
		Steam
W or W/O Reheat	Two pipe, changeover	Fan Coils, Radiation
Low pressure or high pressure	Three pipe	exhaust & infiltration
Single Duct, Reheat	Four pipe	wall apertures
Induction	Terminal Booster Coil	ventilation ducts
Terminal	same as for all air-	low pressure
Low velocity or high velocity	reheat	high pressure
Multi-Path	Steam	vacuum
Dual duct, Variable Volume	One pipe radiation	
reheat	Two pipe radiation	
w/o heat	vacuum	
	low temperature	
	high temperature	
	Terminal Fan Coil Units	
	Terminal Booster Coil	

TABLE 1. TYPICAL ARMY BUILDING HVAC SYSTEMS
(continued)

Unitary Systems

Room Units	Rooftop & Packaged Units	Heat Pumps, Water Loop (semi-central)
Thru wall, windows	fired	
fired	oil, gas	
oil, gas	indirect	
indirect	electric	
electric, heat-pump	steam	
water		
steam		

COST AND PERFORMANCE MODELS FOR RETROFIT AND NEW SYSTEMS

Cost Model

The cost model for HVAC system option evaluation relative to energy use effectiveness requires the incorporation of several standard economic analysis features and also some new ones peculiar to the recent and future behavior of energy costs relative to the general economy. The key feature of the analysis is to properly address, to the best of our ability, the influence of future costs and cost avoidances to fairly weigh current options which will be in use in a future economy. The first and most fundamental aspect of price modelling is to forward project the cost of the project at the time of construction. This requires that current estimates be modified by an appropriate multiplier to reflect the cost anticipated for construction to be completed at the projected date of beneficial occupancy. A project may require three to five years to implement from the time of the recognition of need to the time of occupancy. During this period of time there may be several estimates developed for cost of a comparable project if contracted then. However, the project may not be ready for construction award. The facility engineer may provide one estimate for resource approval, the subsequent architect-engineer designer may, with more detail, provide another estimate, and when finally contracted, after project bid solicitation, result in another cost at occupancy. Therefore, all parties to the estimate procedure must use the same project beneficial occupancy date to have compatible and comparable estimates.

The cost model analysis must reflect numerous other features in addition to use date. Some of these have been addressed previously at some length in DOD publications such as AR 415-17 for Empirical Cost Estimates for Military Construction and Cost Adjustment Factors. The major and non-specific features are discussed briefly in the following remarks. The noted AR contains specific use data. The project cost needs to be normalized or adjusted due to project scale size. Often there are set-up and check-out efforts which are non-proportional to the project scale size. Typically one expects that increased scale size or total cost of the project will influence the unit price, in whatever unit is a convenient measure, i.e., tons of cooling, feet of pipe, MBTUH heating capacity, etc. This is the economy of scale, an assumed or empirical variation of unit cost with size. One word of caution is that as projects get very large, such as in high-rise construction or large scale buildings, the hoped for economy of scale is often lost and increases in unit cost occur. This is due to many features such as the following. As the project grows up or out from a material stockpile, the cost of moving the material via labor and equipment within the project site increases. Also, some features increase in complexity with scale. For example, the air balance and water balance of an HVAC system is inter-related among all of the flow elements in a feedback dependence. Therefore, the complexity increases in a geometric fashion with the scale of project. Although this is usually a minor cost entry in the majority of projects, the trend in its cost is counter to the generally accepted economy of scale philosophy and negates some of the anticipated economy. Also, as projects increase in scale, the amount of inter-trade/craft coordination, supervision and management tends to occur and the unit costs increase.

Project site factors also influence the project cost since wage scales and freight costs to a degree are site specific. The general cost of living and other economic factors influence wages for a given area. The U. S. Department of Labor establishes wage determinations of minimum wages for Federal projects in all regions of the CONUS. In addition, a specific project may be sited away from an area of concentrated labor sources of appropriate skills. This would entail use of financial premiums and other cost elevators such as travel pay, mileage expense and per diem charges to increase costs relative to a site at which required labor is prevalent and nearby. Another site factor of importance in MILCON is the weather influence on productivity of labor. This is obvious for extreme climates such as desert or artic regions but also is significant in temperate climates. Construction of HVAC systems is ordinarily at least partially exterior even in rehabilitation or retrofit projects. Wind, rain, and extreme temperatures will reduce output and limit production. These factors tend to be site specific on the long term so that local experience must be relied upon to judge the influence of the weather on a project. Short-term projects scheduled for a typically bad weather period can be expected to cost more than if it was re-scheduled to a better weather period on the average, given that other factors are not changed. Care must be used in applying general guidelines to project cost estimation that appropriate premiums are included to include weather-site factors or else project over-run from estimate will occur.

In almost all cases of retrofit and rehabilitation, a premium cost will be incurred. This results from two contributing factors peculiar to retrofit. In general, the demolition and removal of existing HVAC equipment and access to the chase areas must be accomplished. This may require removal of ceilings, wall sections, roof openings, and other

architectural disturbances. Also in retrofit, the existing building must be accommodated by new pipe, duct, and wiring runs. Straight line pipe and duct runs may not be possible, and considerable cut and fit and the related labor use will be required. Pipe and duct may be required to run into or through areas of inconvenience which in new construction would be performed in the sequence of construction and before obstructions occurred. Therefore, for these reasons and perhaps others, the retrofit project will generally be more expensive of labor than the new construction project. The additional amount of labor involved depends on the project to a large degree, but the demolition and special fitting required can increase costs by 15-25% of the labor anticipated for comparable new construction. The materials and equipment are not generally different in cost per se for the retrofit project. Since a project in HVAC is usually about 2/3 materials and 1/3 labor, the overall project premium for retrofit over new is about 10% of the composite total costs in many cases for HVAC type systems.

Labor cost adjustment factors also apply to the HVAC projects relative to site specific job difficulty factors. These generally can be lumped into a productivity index relative to unity. These factors include working on second and higher floors above ground, requiring lifting and carrying of material from freight drops. Also, the height above the floor that work is being done influences the cost. Use of ladders, scaffolds and lifts increases labor costs on the job. The compounding of complexity, i.e., second floor work and above 10 feet from finished floor height, should be treated as a multiplication product function. This again would be

compounded by previously discussed weather factors and regional productivity index. Rules of thumb of 10% - 20% labor difficulty factor for work on above ground floors and for above 10 foot work height respectively are generally applicable to labor costs depending upon the height of work above the floor and/or ground level.

Additional factors which apply to cost estimation on materials include that for technological enhancement of equipment during the planning phase which can be incorporated at the construction phase only after added costs. For example, new model machinery may be more energy efficient than that initially selected, but this entails a premimum of cost and could not have been anticipated in the design and planning phase before the improvement was publicized. These technological update factors are particularly important in HVAC controls and flow and regulator developments. Controls costs as a proportion of the total cost of an HVAC project has risen in recent years from about 5% of total cost (materials and labor) to an average 10% of cost for the HVAC system. This is directly related to the recent requirement for off-duty set back, outdoor air temperature resets, improved zoning (smaller zones per building) and general sophistication in minimizing energy use by the system. The development of advanced controls and the Energy Monitoring and Control System (EMCS) auxiliary equipment for HVAC has been rapidly advancing in the past five (5) years and reflects in the cost of controls for HVAC.

Another direct influence on HVAC system cost is that related to improved efficiency independent of technology improvement but resulting from the use of better quality or more efficient components. For example, duct and vent dampers of conventional or nominal quality may leak up to 5% or more in

the closed position. High quality dampers with 1% leakage are initially more expensive, but, in most instances, are now justified to conserve air flow energy and to reduce parasitic losses of conditioned air. Other examples relate to operational efficiencies of specific components such as pumps and refrigeration compressors which are selected by capacity to match best operating points. Often a given capacity is not a stock item and a larger (or smaller) capacity is selected which in the application of interest does not perform as well as would the proper size. There may be a premium cost to acquire the required size or a delay in delivery. It must be recognized that these off-nominal design selections can create life-long energy waste and related costs. Therefore, although there is no literal technology improvement, proper acquisition of best fit equipment to each application is important.

Approximate Estimation of Fuel Consumption for Savings Evaluation

In the heating season, the fuel consumption or energy use of the facility will vary with the seasonal severity relative to the design point conditions. This effect is usually accommodated by converting the seasonal demand to a degree day equivalent. The maximum and minimum recorded dry bulb temperatures of each day are averaged and the constant 65^{o} F is subtracted. The resulting difference is the degree days of heating required relative to a 65^{o} F average. The same function applies to summer also, except that the degree days are for cooling. The use in summer conditions is not as appropriate as winter because humidity and cloudiness influence cooling demand significantly. The degree days for an average

heating season at a given location is an historical record prepared by the weather service and often by utility companies. Weather data is also available from the U. S. Government Printing Office <u>Engineering Weather Data,</u> Air Force, Army and Navy Manual for Facility Design and Construction (1967). For a given Army base, the local administration may have recording equipment to monitor this factor. The average energy use per heating season is estimated for rough purposes by the product of installed capacity per differential degree (indoor-outdoor) and the annual degree days. The resultant input of annual heat energy is:

$$E_u \, (10^6 \text{BTU}) = \frac{Q_D \, (\text{D.D.})}{(\Delta T)_D} \quad \frac{24}{10^6}$$

The Q_D is the peak demand installed capacity in BTU per hour, D.D. is the local annual degree days, and ΔT_D is the design difference between indoor and outdoor dry bulb temperature in ^oF at peak heating demand.

This approximation does not accommodate night set-back or varying ventilation when unoccupied. However, it is an estimator to determine if a given system choice is cost effective; for example, if installed capacity at design point can be reduced. To obtain the estimated total cost of heat, use the fuel cost per 10^6 BTU of the fuel used to provide heat. In using this value, it is necessary to accommodate future rate inflation factors for FY79-82 provided in Table 2 of ECIP Guidance Memo from the Deputy Assistant Secretary of Defense of 21 October 1977,

as repeated below.

	FY79	FY80	FY81	FY82
Coal	10.0%	10.0%	10.0%	10.0%
Fuel Oil	16.0%	16.0%	14.0%	14.0%
Electricity	16.0%	16.0%	13.0%	13.0%
Natural Gas & LPG	15.0%	15.0%	14.0%	14.0%

For long term life cycle estimates of fuel cost, the differential escalation rates provided in the same reference may be used as below.

Coal	5.0%
Fuel Oil	8.0%
Natural Gas & LPG	8.0%
Electricity	7.0%

The life cycle cost of fuel energy in a building system is estimated by using the current fuel energy cost and applying the discounted, differential inflation rate factor provided in the noted ECIP memo for the economic life of the building (years) and the specific fuel differential rate stated above.

The annual savings in fuel, or the lifetime savings in fuel, due to applying an off-hours zone temperature setback can be estimated from information developed by the Federal Energy Administration for set-back conditions as a function of the local climate and occupancy levels. The following Figure 1 illustrates savings to be expected in a 60-hour per week occupancy situation, relative to no setback and 70^0 room conditions. It should be noted that percent improvements at high seasonal degree-day demands are low. However, from the previous discussion where annual heat

energy use is shown to be approximately proportional to the degree days, the value of the energy savings may be significant, even at low savings percentages. Note also from Figure 1 that higher setbacks yield diminished savings per degree. The data from the estimated savings chart and from the total annual heat energy estimate will provide the annual, or lifetime, estimate of utility energy with and without winter setback conditions.

Examination of the chart reveals some rather obvious conclusions. In mild climates (2000 degree-days) a 10^{o} setback provides a substantial (48%) heat energy savings because the outdoor average temperature only seldom drops substantially below 55^{o}. In general, unoccupied period setback controls will always prove to be cost effective because they are so inexpensive relative to the lifetime energy cost savings. In severe climates, lower percentages are achieved but the total quantities are significant. For example, consider an 8000 degree-day design temperature difference and a 10^6 BTU per hour peak output capability. Annual energy estimate for heating is, per the previous discussion, equal approximately 2.4×10^9 BTU. For 60-hour per week occupancy and a 15^{o} setback in unoccupied periods, the savings is 40% per year or about 10^9 BTU. At current prices of about \$3 to \$4 per 10^6 BTU of oil heat, the savings per year is about \$3000 to \$4000.

In applying the set-back principle, care must be taken to realize the ramifications. It may become necessary to increase the peak heating capacity from that for a steady state design condition, because, with setback, it is necessary to overcome the transient as well. Therefore, in using the annual energy estimator, care must be exercised that transient dictated

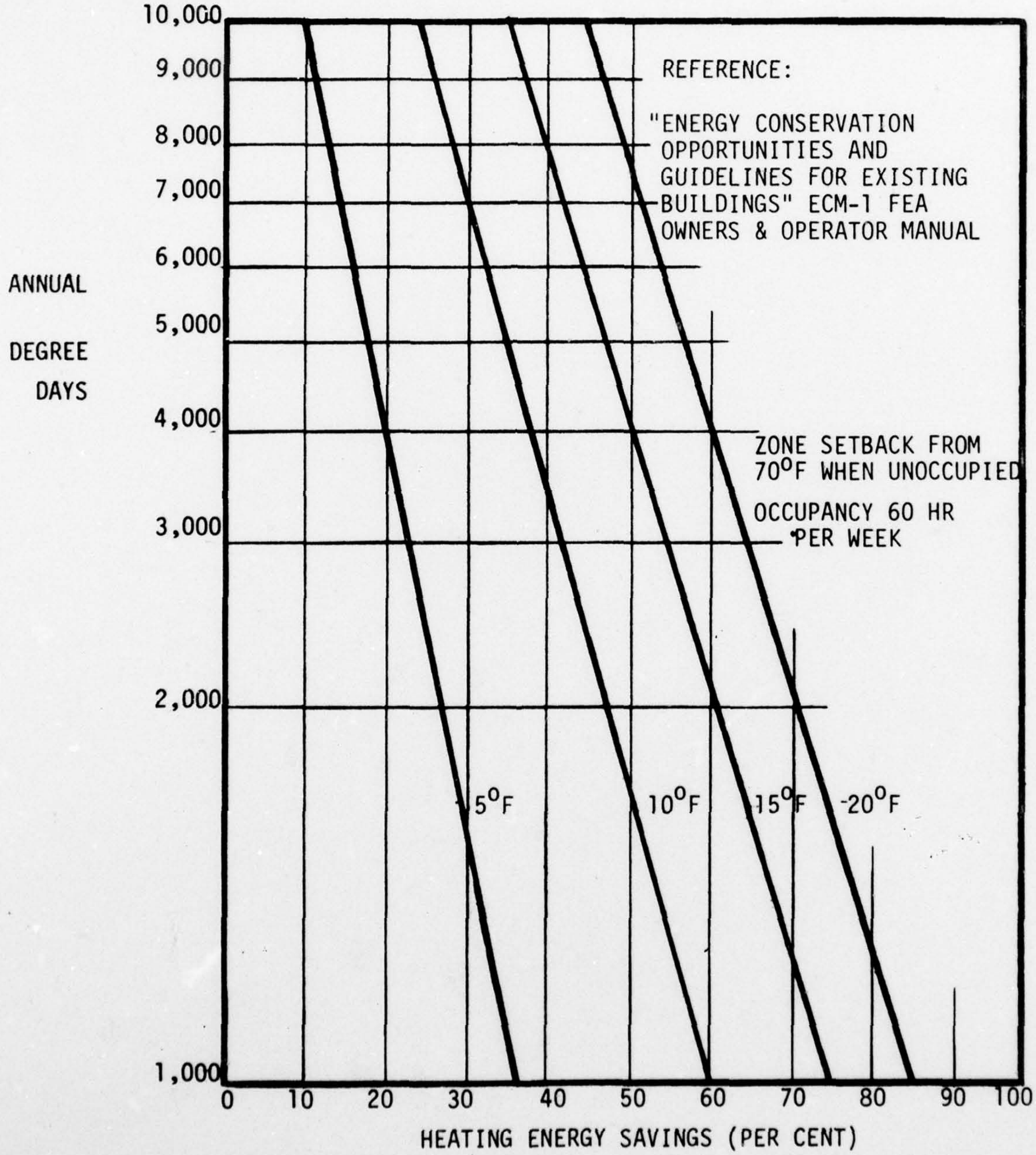
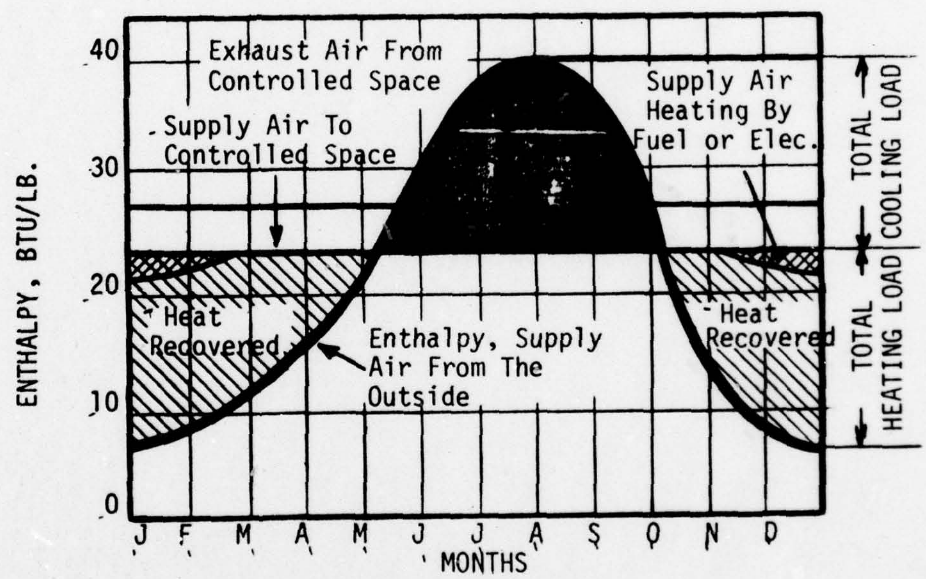


Figure 1. Estimated Annual Savings for Unoccupied Setback in Winter.

overcapacity in Q_D is not included in the estimate to eliminate inaccurate projection. However, the system will have to be oversized from the steady design condition value, and first cost will increase with this over capacity effect. As deeper setbacks are elected, the magnitude of this overcapacity will increase. Therefore, due to diminishing energy savings per extra degree of setback and also the cost of increased capacity for heating to overcome the transient condition, there will result a most cost effective setback for each facility, climate, and occupancy level. For the above noted example, consider that a 30% overcapacity is dictated to achieve reasonable transient response at 15^{o} setback. The overcapacity of 300,000 BTU per hour may cost an additional $10,000 in furnace, pipe, duct, and equipment costs, thereby reducing the life cycle cost benefit. Another effect that is not included as yet, but which reduces the potential demand for overcapacity, is ventilation energy and flow controls. Steady state peak demand usually includes a factor for heating ventilation make-up air. With proper sequencing, the warm-up period can be done while unoccupied and at greatly reduced ventilation flows in general. Therefore, the required overcapacity can be less than necessary with ventilation heat loss effects. Also, the ventilation air system can be fitted with energy recovery equipment exchanging with the exhaust to reduce the installed capacity from that which would have been selected without such recovery equipment. In essence, a system designed to current higher ventilation capability standards and without heat recovery represents an overcapacity available to accommodate set-back transients if such conservation controls and equipment are added as a retrofit. A graphical example of the heating and cooling required to condition ventilation air with and without enthalpy recovery equipment of typical operating efficiency is presented in the following Figure 2.



Solidly shaded areas show the heat recovery and cooling reduction by an air-to-air enthalpy heat exchanger operating on a year-round basis. Crosshatched areas signify need for additional cooling and heating.

Figure 2. Air-to-Air Enthalpy Heat Exchanger Heat Recovery and Cooling Reduction.

Enthalpy recovery includes both sensible and latent heat exchange, and, therefore, is the most advanced recovery system.

The relationship of the heat recovered to accommodate ventilating air and that required for conduction losses, etc., is not specified since each facility has different ventilation requirements in CFM per square foot of space. Note that over a typical year's period the enthalpy recovery system provides the majority of energy required for ventilation conditioning. In this case, the heating requirement is reduced to 6% of the non-recovery load and air conditioning (cooling and dehumidification) to 49% of the non-recovery load. Energy used to operate the recovery system is not included in the above savings. In the future, using setbacks and using ventilation controls during non-occupancy and also heat recovery, the peak heating and cooling load design point capacity may be larger, smaller, or the same as for current and past criteria designs without such equipment. This is because the selected setback, the climate, and the ventilation flows all influence the capacity choice.

In summary, the amount or cost of heating energy required for an HVAC system in a given climatic environment can be estimated from readily available locality data and the nominal heating capacity for meeting peak steady state heating requirements. This estimate is useful for evaluating the energy savings associated with setback and ventilation controls incorporated in retrofit construction. The installation of increased capacity to meet transient recovery from setback is an increased first cost that must be accommodated in life cycle costing estimates. It is pointed out that ventilation controls and heat recovery equipment both act to reduce this peaking effect in transients, but the net change in

recommended capacity over current practice is not a simple generalization because of varying climate, setback, and occupancy requirements, as well as ventilation demands peculiar to the building.

Air conditioning savings from setback of temperature is less direct than that for the heating since humidity, i.e., wet-bulb temperature, plays a more significant role in the summer conditions than it does in winter in most applications. Degree days of dry-bulb difference from 65^{0} is readily measurable and retrieved from records. However, wet-bulb temperature records are not as prevalently available. One measure of the energy used in summer air conditioning is the average hours of annual operation of the system at rated capacity for the locality and building use. Continuous use systems such as dual duct units with continuous low levels of air conditioning demand are more difficult to characterize with this method. The ASHRAE recommended range of equivalent full-time operation for cooling, for energy consumption estimates are presented in Table 3 for numerous CONUS locations. For ready reference, the typically achievable performance on electrical consumption per refrigerating ton (12000 BTU per hour cooling) is presented in the subsequent Table 4. The data is very approximate and dependent upon numerous unstated parameters. For example, the energy consumption per ton is also dependent upon condenser temperature which typically rises in the hottest outdoor conditions to reduce efficiency or C.O.P. (coefficient of performance). Also, the zone set-point in temperature affects performance due to the potential variation in evaporator temperature and of increased heat transfer effectiveness. This data is from the ASHRAE 1976 Handbook and Product Directory. Note that the efficiency or performance of the larger

TABLE 3. ESTIMATED EQUIVALENT RATED FULL-LOAD HOURS OF OPERATION FOR PROPERLY SIZED EQUIPMENT DURING NORMAL COOLING SEASON

Albuquerque, NM	800-2200	Indianapolis, IN	600-1600
Atlantic City, NJ	500-800	Little Rock, AR	1400-2400
Birmingham, AL	1200-2200	Minneapolis, MN	400-800
Boston, MA	400-1200	New Orleans, LA	1400-2800
Burlington, VT	200-600	New York, NY	500-1000
Charlotte, NC	700-1100	Newark, NJ	400-900
Chicago, IL	500-1000	Oklahoma City, OK	1100-2000
Cleveland, OH	400-800	Pittsburgh, PA	900-1200
Cincinnati, OH	1000-1500	Rapid City, SD	800-1000
Columbia, SC	1200-1400	St. Joseph, MO	1000-1600
Corpus Christi, TX	2000-2500	St. Petersburg, FL	1500-2700
Dallas, TX	1200-1600	San Diego, CA	800-1700
Denver, CO	400-800	Savannah, GA	1200-1400
Des Moines, IA	600-1000	Seattle, WA	400-1200
Detroit, MI	700-1000	Syracuse, NY	200-1000
Duluth, MN	300-500	Trenton, NJ	800-1000
El Paso, TX	1000-1400	Tulsa, OK	1500-2200
Honolulu, HI	1500-3500	Washington, DC	700-1200

TABLE 4. APPROXIMATE POWER INPUTS

System	Compressor Kw/Design Ton	Auxiliaries Kw/Design Ton
Window Units	1.46	0.32
Through-Wall Units	1.64	0.30
Dwelling Unit, Central Air-Cooled	1.49	0.14
Central, Group, or Bldg. Cooling Plants		
(3 to 25 tons) Air-Cooled	1.20	0.20
(25 to 100 tons) Air-Cooled	1.18	0.21
(25 to 100 tons) Water-Cooled	0.94	0.17
(Over 100 tons) Water-Cooled	0.79	0.20

units, even with the required auxiliaries is substantially better than the window units and small packaged coolers. Current practice is changing to stress the reporting of performance as an electrical efficiency ratio or E.E.R. which is the BTU delivered as cooling per watt-hour of electrical input. To convert the above mentioned KW per ton use the following:

$$\text{E.E.R. (BTU/watt hour)} = \frac{12.0}{(\text{KW/Ton})}$$

Note that the range of the E.E.R. is almost from 6.0 to 12.0 depending on size, etc. Knowing the cost of electricity per KWhr allows estimation of summer cooling cost for energy savings evaluation purposes. Care must be exercised to include any fossil fuel adjustments or quantity use premiums for peak demand that apply in many cases.

Another aspect that impacts system operating cost for cooling, but not directly the energy consumption, is the so called demand charges which are prevalent in public utilities. This is a charge applied, usually on a sliding scale, based upon the peak power (kw) demand made by the user for a specified short averaged period (5-30 minutes). This charge reflects the utilities cost of providing large transmission and generating capacity to meet peak demands which are in excess of average consumption. The ratio of average demand (kw hour per hour) and peak demand is referred to as load factor (L.F.). In commercial and industrial applications, the demand charges may represent 40% of the total billing. In conducting life cycle cost analysis, the savings in demand charge resulting from lower peak demand due to higher efficiency, less ventilation, set-back, etc., are tangible savings if a demand schedule of rates is in effect. It appears that, in the future, the public electric and gas utility industry is moving toward demand rate schedules as well as consumption charges. The higher first cost, financing, lead time, environmental controls and regulatory restrictions on electric power plant development and gas storage and distribution systems cause the utilities to vary energy price with peak demand. A premium cost is incurred. A facility HVAC system can accrue a cost benefit at no savings in actual energy if it is advantageously designed to avoid peak demands in electricity. Since natural gas fuel will only be used in limited future new installations and renovations, the demand charge avoidance on gas consumption will not enter too often in life cycle cost studies. Neglecting heat pumps and electric resistance heaters as significant general options for Army buildings for the moment, then electric peak demands can only be reduced in a limited number of ways for typical HVAC systems, namely:

1.) using higher efficiency motors to reduce input-output mechanical energy losses;

2.) using higher efficiency machinery elements such as fans, compressors, pumps;

3.) using design choices which minimize the ideal requirement for transport and distribution energy; for example, using low pressure, low velocity air duct, or using larger flow pipe to reduce pumping functions;

4.) using control logic which avoids sharp peaking demands to meet transients; for example, using timer anticipation to recover from setback over a longer period to avoid installing significantly over-capacity equipment;

5.) using ventilation energy recovery to reduce peak capacity of installed equipment;

6.) incorporating feedback controls in HVAC equipment to seek optimal thermodynamic operation; for example, programming chiller water outlet temperature to respond to ambient air enthalpy and the return air conditions to demand least horsepower per ton of cooling for prevailing conditions.

The demand charge effects may be further complicated in some areas of the CONUS by time-of-day premiums and season-of-year premiums. For example, a utility company in a region of the country where peak demands are experienced in the summer cooling season would apply premium cost to mid-afternoon in summer demand peaks. Another region may experience peak demand in winter due to electric heating use. The premium on demand for large consumers may be as much as $6.00 per month per peak kw.

Therefore, in an area where consumption costs are \$.02 per kw hr, a short-term peak use item (two hours of peak use, 30 days per month) will incur a 120 kw hr per month (\$2.40) consumption charge per kw, but up to \$6.00 demand charge. Therefore, the effective rate of the utility for this specific item would be \$0.07 per kw hr actually used.

A very approximate estimate of the reduction of air conditioning energy savings in summer resulting from setback can be intuitively derived. During summer unoccupied periods, the setback temperature would actually be an increase over the occupied period temperature setpoint. An upper limit on savings would be to reduce the average annual cooling operation hours by the hours of facility non-use. However, this will always be an over estimate of savings because the HVAC system will be called on to a.) sustain the setback (up) temperature in off duty times, and b.) to recool/dehumidify on re-occupancy. To make a first order approximate correction to the noted reduction due to non-occupancy, consider the following. Rule of thumb design practice is to provide for approximately $\Delta T = 15\text{-}17^{o}$. Therefore, the non-occupancy periods will operate at most at 2/3 of capacity. This neglects the fact that the enthalpy ratio including latent heat may not be equal the temperature ratio of dry bulb values. The recooling on re-occupancy depends upon relative capacity but a value of one hour at full capacity is probably typical of the extra cooling required. Unoccupied periods will generally incur less lighting load, less equipment load, and obviously less people based load. Also, most unoccupied situations are at night with no solar load. Therefore, the unoccupied period cooling savings is only a fraction of the equivalent of time compared with daytime operation. For example, in a 12-week cooling period, at 5^{o} set-up and 14 hours per

day non-occupied period on seven days per week, with daily set-up/back the reduced cooling time is approximately 140 hours. Therefore, in a season with 1000 anticipated full capacity cooling hours of operation, a 14% savings in cooling energy would be predicted by the noted 5^{0} setback example. Verification of this trend will require an analysis of facility records for typical buildings.

The reduction of cooling operation in off-duty periods depends upon the unoccupied setback temperature and number of cycles per season. In an example calculation for a 12 week cooling season 10 hour per day occupancy, 7 days per week and a 5^{0} setback, the 1000 hour anticipated cooling hours of operation was reduced to 860 hours.

Performance Model

Insulation

Performance models for retrofit and new construction HVAC equipment are difficult to generalize because of the extremes of design features that prevail in the field. Specific recommendations are possible when the HVAC function is identified. The energy loss due to incomplete insulation and the excess over the ideal of energy used to operate the HVAC system are specific areas that may be treated. Areas conducive to performance modelling include insulation, duct sizing and pipe sizing. For years the most economical selection of insulation for pipe and duct has been a common topic for analysis. The advent of computers has led to the use of specialized codes to compute the best choice of insulation for an application. These codes rely on the physical fact that increased insulation will reduce energy loss (a life cost savings) but will require added initial expenses in material and labor. The Thermal Insulation Manufacturer's Association has developed a publicly available prediction system for pipe and duct. In conducting a performance model analysis, it is necessary to recognize that the excessive use of added insulation is self defeating because the added bulk will increase the ultimate outside envelope area and, therefore, the energy loss will actually begin to increase over that of a minimum loss insulation size. In addition, the material and labor cost will generally increase with added insulation while energy loss cost recovery diminishes with incremental insulation additions up to the point where losses again increase. One way of displaying the effect of insulation thickness choice on energy loss is presented in the following Figure 3.

In this figure, the percent energy loss per 100 feet of duct run is plotted versus duct capacity (SCFM) for typical applications. The percent energy loss is in the ratio of the air energy loss in the 100 foot run to the energy conducted by the differential temperature between the duct interior and the ambient air. The size of duct is not of particular influence because the practical velocity of air duct flow ranges over a relatively narrow band, and at increased velocity (high face velocity) heat convection effects are significant while at low face velocity they are not as significant. However, high face velocity implies small perimeter for a given flow rate (SCFM) and, therefore, small outer perimeter heat losses. Note that in Figure 3 the energy loss per 100 feet of duct rises sharply for low capacity, i.e., less than about 3000 CFM duct runs, independent of the amount of insulation present. It becomes obvious that small flow runs should be kept to a minimum and the main trunk relied upon as much as possible to transport the air supply. The relative advantage of 2 inch insulation thickness over 1 inch is shown to be almost a constant 1% per 100 feet above about 4000 CFM capacity and about 2% per 100 feet for lower capacity duct. Losses for uninsulated steel are so great as to preclude the practical consideration of bare duct. Every 1 inch of insulation will reduce loss about 5 fold over bare duct. To conduct an insulation and duct economic analysis, the annual energy flow through the duct can be estimated from building use data and records and the duct design capacity layout. Note that in retrofit, energy can be saved by better use of main trunk duct

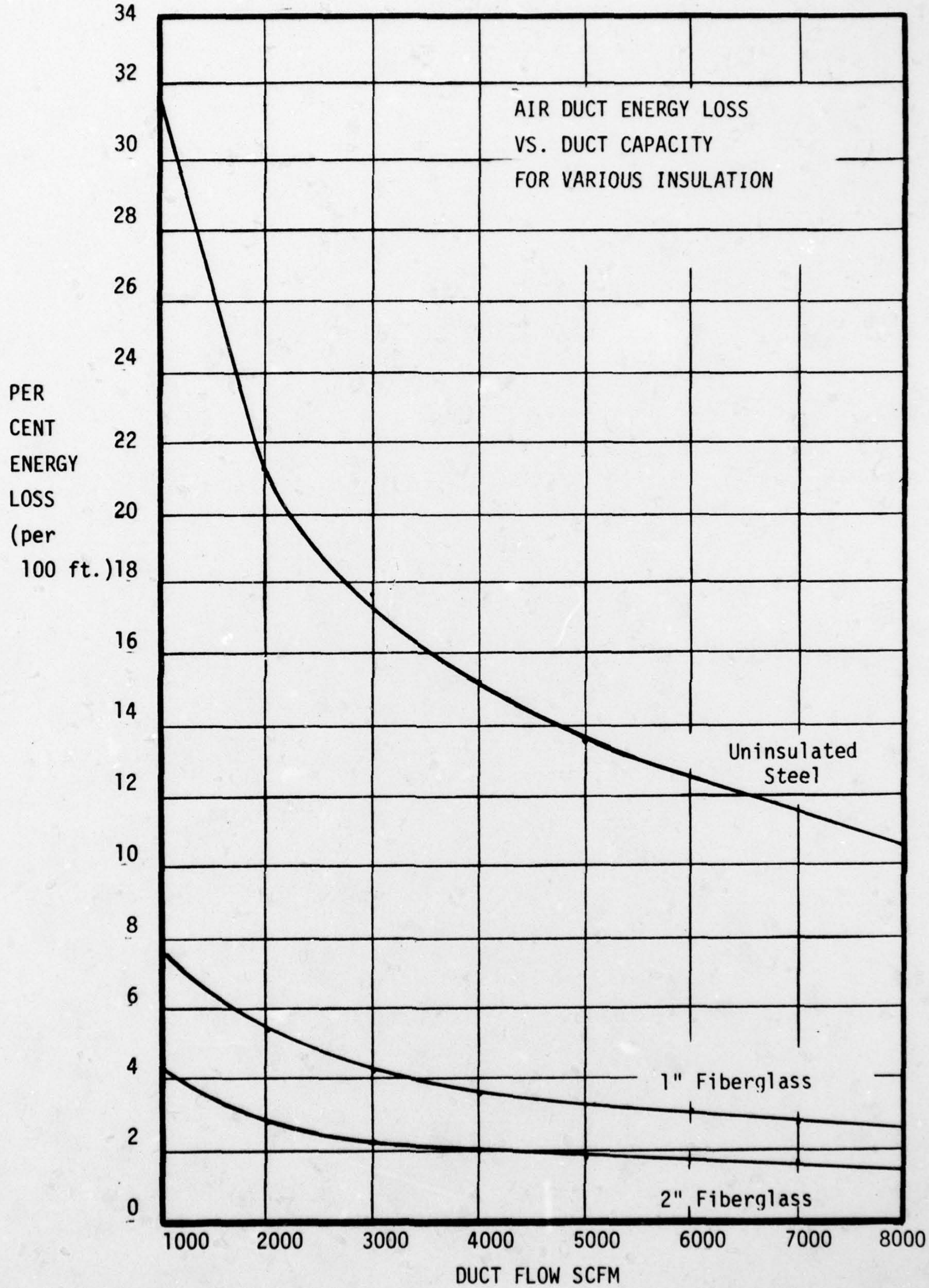


Figure 3. Air Duct Energy Loss.

and reduced lateral runs, as well as by increasing insulation. The above noted general results are applicable to typical duct geometrical shapes and average turns and fittings. Extreme shapes of very flat duct for tight clearance applications, outside of ASHRAE accepted recommendations for duct aspect ratio and carrying capacity, will depart from the behavior noted above. The result of the energy analysis and typical current cost of insulation material and labor is such that the payback in MBTUY per $K ratio is about 69 for 2 inches on a 5000 CFM duct capacity in steady use year around, and proportionately less for lower use levels. This is relative to 1 inch of insulation. However, this is the incremental cost over 1 inch thickness for the second inch when retrofitting and replacing the whole insulation. To add a second inch of insulation to an existing inch of insulation when not part of a general retrofit, i.e., removal of all old insulation, the ratio is only about 20, and, therefore, marginally worthwhile. There are other projects of greater economic worth. In summary, for air duct situations, the most economical arrangement is to have large capacity trunk ducts which have minimal lateral duct, low capacity, duct runs because these low capacity ducts with high perimeter to area of flow ratio lose 2 to 3 times the energy per CFM as the large ducts.

The most economic insulation thickness is, for most cases, found to be in the 1 1/2 to 2 inch thickness range, and either of the mentioned sizes could be selected. The lower size would be appropriate for short economic life buildings, and the larger for larger term building life projections or to build in a greater hedge against future energy rate inflation. In a broad

optimum such as this, the selection of a point of design on the over-thick side of insulation will result in less energy loss, therefore, any slight errors in under capacity of equipment, etc., will be compensated for. Also, as noted, the rather uncertain future energy prices will be accommodated, since the optimum design thickness will shift to the side of thicker insulation as energy rates escalate further than planned. The usual case is that the true optimal thickness is not a stock size and, therefore, a choice of off-optimum is necessary. The installation labor and outer wrap, hangers, etc., are not very dependent on insulation thickness and, therefore, the only real cost is for material of the insulation itself, which often is not in direct proportion to thickness, i.e., an economy of scale exists.

Pumping Power and Pipe Size Selection

The current and future energy economy situation is such that design criteria must be changed from past experiences and recommendations. Pipe sizing for fluid carrying requirements for HVAC systems, i.e., hot and chilled water, condensate, etc., were usually sized on criteria or pressure drop and pipe cost as applied to past energy cost schedules. In the future, the rising energy costs are such that continuous or near continuous running equipment will consume a cost of energy in its lifetime just to move the fluid which is in excess of the cost of an additional pipe diameter selection for lower losses. An economic analysis coupled with a pumping power analysis was conducted to arrive at the cost-optimal pipe size to minimize cost and to reduce energy of pumping. It was found that the least cost piping and pumping system resulted from pipe diameter selection about

1.4 times the value that would have been selected in an era of lower energy costs and stable equipment and material costs. An interesting note is that the resultant pumping energy for the same function is only about 0.22 that of the prior design criteria. Therefore, the choice of the next largest pipe size to represent the 1.4 factor will result in a minimum life cycle cost and a significant lifetime pumping energy reduction. A further analysis of sensitivity to changes indicated that further selection of an even larger than cost optimum pipe size would reduce life time energy to about 0.15 of the current (past) criteria. The further reduction of life energy from 0.22 to 0.15 of current can be achieved at very little increase in cost from that of the minimal life cycle cost. This is explainable from several intuitive as well as analytical arguments. At larger pipe sizes, as are used to reduce energy, the labor, hangers, pipe material and insulation do not increase in cost as rapidly as for small sizes. Also, as in most physical systems analysis, there are no sharp discontinuities in cost, just gradual variations. The result of this argument is that a selection of a pipe size of 1.4 to 1.6 times the diameter previously selected for the same flow rate is economically justified. This will increase initial cost, all other things being equal, but will reduce life cycle cost to near minumum.

Another feature affecting pipe size cost optimization and energy use is the flow rate per unit of energy transmission from the fluid. Contemporary practice based on old economics tends to use minimal fluid temperature drop or use and recycle large quantities of material. Typical water cooling drops are about 8 to 12 degrees Fahrenheit and in heating about 20 degrees Fahrenheit for water. These decisions are based on:

a) older, less heat transfer efficient terminal units, and b) the usual first cost economy of using smaller, less efficient terminal units for heat transfer. Due to heat transfer principles too lengthy to reproduce here, the use of higher temperature drops can be balanced with choices of increased terminal system heat area and efficiency to achieve the same gross effect at flow rates about 1/2 to 2/3 of the current practice based on low cost electrical (pumping) energy. The combination of the previously noted increase in pipe size for the same flow rate to reduce pump power and the reduction of flow rate to reduce fluid cycling rate allows the use of the same or nearly the same pipe size as predicted from current practice but with about 1/2 to 1/3 less flow rate. Therefore an optimal cost and energy life cycle can be achieved with current pipe size criteria and installations. The essential recommendation of doubling pipe transverse area (diameter ratio $\sqrt{2}$) is offset by reducing flow by 1/2 due to use of higher temperature drops in terminal equipment. Therefore by using higher efficiency terminal equipment about an 80% reduction in pumping energy over the life of the piping system is achievable. It is not necessary to change design guides, handbooks or other source materials based on terminal unit capacity to achieve these energy savings. It is necessary only to recognize that the flow through terminal equipment will be about half that of current practice and it may require small variations in temperature profiles, i.e., use of 190^{o} water vs. 180^{o} water to achieve heating in a given terminal unit at the reduced flow rate. The current and future availability of improved heat transfer effectiveness equipment through the use of advanced concepts in extended area surfaces will permit use of lower temperature supplies to provide the same output.

EVALUATION SELECTION AND DISCUSSION OF OPTIONS

Energy Savings from Equipment Modifications

Considerable operating energy can be saved with modest cost in typical HVAC systems. In this section, several items are addressed which are specific cost effective options.

<u>Damper and Duct Leakage</u>

An air distribution system for the HVAC operations will consist of a supply and return duct system with dampers on; outside air supply, exhaust vents, and between the hot and cold supply air. The uncontrolled flow of controlled, conditioned air out of the system to the uncontrolled space or outdoors through leakage of ducts at joints or at leaking dampers is an energy loss. In winter, or at night, outside air dampers may be nominally closed, but leak a substantial amount due to poor sealing. Ducts improperly joined or damaged will leak air also. The energy cost of these leakages can be estimated in a fashion similar to that used in the annual vent air energy loss since it is the same application. The cross-leakage of air from the cold duct to the hot duct is slightly different in analysis.

Consider a case in which the leakage loss induces outdoor air make-up to the circulation system such as may occur at inlet and discharge dampers resulting from damper leakage or duct leakage. The energy lost due to a loss of conditioned air in the fraction (f) of the air handling capacity is given as follows from the prior analysis on vent loss.

$$E \text{ (annual BTU/CFM airflow)} = ft \left[W_H \times 1.08 \; (T_i - T_o) + W_c \times 4.5 \; \Delta H_c \right]$$

where f is the fraction of leakage, t is the hours per week that the air handler system operates in the leakage condition. The other elements of

the function are provided for typical cities in the CONUS in Table (5) charts. A typical high quality damper will leak approximately 1% of the full-open flow when shut-off and under the same pressure difference. A poor quality or damaged one may leak 10 to 15% of full-flow. Therefore in using a control strategy with unoccpied period outdoor air shut-off, the expected savings may not materialize due to leakage at dampers and vents. The loss of energy due to duct loss/leaks can also be in the 5-15% range due to several causes:

a) loose fitting or missing covers on duct heaters, air handlers, reheat coil mounts, etc. where pipes, tubes, wires, etc. penetrate the ducts.

b) leakage at deteriorated boots and flexible fittings connecting air handlers to duct.

c) leakage at joints in the duct and at junctions.

The annual energy loss due to duct leakage of hot and cold duct flows through leaking dampers or mixing boxes in dual duct and multi-zone systems is more difficult to estimate but can be approximated as follows. The energy required to chill air from the average room conditioned state to the supply mixed air condition is negated by the energy provided by the heating system to heat air in the hot duct. As the two streams mix the mixed flow does not contribute to conditioning the controlled space. If the hot duct supply is $20^{o}F$ above the controlled zone temperature and the cold duct is $20^{o}F$ below the controlled zone temperature, then neglecting the latent heat effects, the air quantity which mixes to the room condition by virtue of the leakage does no appreciable good to meet HVAC demand

TABLE 5. HEATING AND COOLING SEASONS
BY LOCATION

	Average Heating Temp	Heating Season Length in Weeks	Cooling Season Length in Weeks	Average Cooling Season Temp	BTU Per Pound ΔH_c
Albany, NY	39°F	33	19	75°F	5.1
Albuquerque, NM	45°F	25	27	80°F	.0
Atlanta, GA	48°F	22	30	79°F	7.5
Bismarck, ND	35°F	34	18	78°F	.8
Boise, ID	42°F	32	20	78°F	.8
Boston, MA	40°F	33	19	75°F	4.3
Billings, MT	41°F	34	18	77°F	.4
Buffalo, NY	39°F	33	19	74°F	5.3
Charleston, SC	51°F	16	36	79°F	9.1
Chicago, IL	38°F	31	21	76°F	5.7
Corpus Christi, TX	56°F	9	43	81°F	13.0
Dallas, TX	49°F	18	34	83°F	8.2
Denver, CO	42°F	29	23	77°F	.0
Detroit, MI	38°F	33	19	75°F	5.0
Ellsworth, SD	39°F	32	20	77°F	.9
Fairchild, WA	40°F	36	16	75°F	.1
Greensboro, NC	46°F	24	28	79°F	6.0
Helena, MT	38°F	37	15	75°F	.0
Kansas City, MO	42°F	26	26	80°F	6.4
Kodiak, AK	40°F	51	1	67°F	.5
Las Vegas, NV	50°F	17	35	88°F	2.1
Los Angeles, CA	57°F	19	33	71°F	2.4
Louisville, KY	44°F	26	26	80°F	7.8
Lubbock, TX	46°F	21	31	80°F	2.2
Memphis, TN	47°F	22	30	80°F	9.6
Miami, FL	58°F	2	50	80°F	12.5
Minneapolis, MN	35°F	33	19	76°F	4.7
New Orleans, LA	54°F	12	40	80°F	12.1
Omaha, NE	38°F	29	23	78°F	5.7
Pearl Harbor, HI	62°F	0	52	80°F	10.3
Phoenix, AZ	53°F	11	41	87°F	3.9
Pittsburgh, PA	41°F	30	22	75°F	5.0
Portland, ME	40°F	37	15	73°F	3.5
Portland, OR	49°F	36	16	72°F	1.6
Roosevelt Rds., PR	62°F	0	52	83°F	15.1
Sacramento, CA	49°F	20	32	84°F	1.8
Salt Lake City, UT	40°F	32	20	79°F	.0
San Diego, CA	58°F	22	30	70°F	2.7
San Francisco, CA	52°F	28	24	71°F	6.7
Traverse City, MI	37°F	35	17	74°F	4.7
Tulsa, OK	45°F	22	30	81°F	7.1
Washington, DC	44°F	28	24	77°F	6.4

except to contribute to volume flow. The heating and cooling sources must operate at increased output to meet demand.

For the above example, the total energy lost is given by:

$$E \ \frac{\text{Annual BTU}}{\text{CFM Capacity}} = 52Lt\left[1.08\left(20 + \frac{20}{cop}\right)\right]$$

The (L) is the leakage as a fraction of air handler capacity and t is the operational time per week in hours that the system leaks flows. The cop term is the coefficient of performance of the chiller system indicating the actual energy consumed in providing cooling. This is specific to a cooling system design and performance but for this approximate analysis a value of 2.0 is probably adequate when considering auxiliary equipment parasitic energy losses to provide the cooling. Therefore the winter loss for a full time system is for a 26 week heating season.

$$E \ \frac{\text{Annual BTU}}{\text{CFM Capacity}} = 1415 \text{ per \% crossover leakage}$$

Therefore if 5% of the flow is cross-over leakage and not controlled, the annual energy loss per CFM is 7,075 BTU. In a 10,000 CFM circulation system capacity this is about 70×10^6 BTU per year. At current prices of energy as noted in prior sections this is equal to \$400 to \$800 per year, neglecting inflation of fuel prices. The diffuser and damper leakage as noted can be a considerable energy waste in the dual duct and multizone approaches to HVAC. In another section of this report the loss due to over-ventilation in non-occupied periods is examined and discussed. In general, the vent damper will be closed during non-occupied periods and be subject to leakage conditions. The leakage which is outdoor air make-up either at the damper or by infiltration due to lowered

pressure will create a cost of energy due to the waste. The analysis is the same as for the previous example except that when ventilation is assumed to be stopped, the actual savings will not meet anticipation unless dampers are low leakage and other sources of outdoor air are effectively closed.

Unoccupied Space Temperature Reduction

Night set-back of temperature in non-occupied space has been discussed in earlier reports and found to be a good approach to savings of energy and cost at minimal investment. Also, the reduction of temperature in non-occupied zones of a facility which is active in other zones is also worthwhile. This has the prospect of saving ventilation air energy and also the transmission energy loss through the walls. Installing zone dampers and operating motors on vent air and zone thermostats on the zone heat if separate will allow set-back and reduced venting of non-occupied areas such as conference rooms, non-occupied classrooms, etc. within an otherwise occupied facility. An estimate of the potential savings is possible from the two sources of loss, i.e. vent air and transmission.

The average thermal transmission coefficient for the construction taking into account the window effects must be known or estimated. Typical values vary from 0.5 to 2.0 in BTU/sq. ft./hr/^{o}F. A 10^{o}F set-back from the normal set point will save from 5 to 20 BTU per hour per square foot of exposed wall of the zone, assuming that outdoor temperature is not included in the range of normal to normal less 10. A 10 x 10 x 10 room with one exposed wall and thermal transmission of 1.0 will save

1,000 BTUH in energy with only the $10^{o}F$ set-back compared with normal constantly occupied comfort setting. The vent air at 0.1 CFM per square foot for the same zone would result in 10 CFM vent air savings in unoccupancy for a savings of about 270 BTUH for an average ΔH indoor to outdoor condition of 6 BTU/lb. Therefore the total savings for the example room zone is about 1270 BTU for every non-occupied hour that the zone is 10^{o} less than the normal set-point.

Boiler Water Temperature Reset

In hydraulic systems the circulated water temperature can be programmed to match the demand. In most applications, the water temperature is scheduled in inverse relationship to outdoor temperature. This accomplishes several desirable features. The full flow rate of water is used in most units as at the peak heating demand point. Therefore metering of flow and the accommodation in the control logic is minimized. This reduces over-heating situations from occurring in specific zones. Also, the boiler or converter can operate at higher efficiency because the water temperature at which heat is circulated is lower. The vent-stack loss can be less, and the loss due to conduction through insulation is less as well when the water temperature is lower. The water temperature only matches the high limit demand of the system at design point conditions which occur only infrequently. The savings that can be achieved by boiler water reset are difficult to quantify without a detail analysis of each application. The reset can be done manually or by an automatic outdoor temperature sensing valve to set water temperature. By monitoring the outdoor temperature and applying reset to optimum conditions,

as much as 15% heating savings are possible in Army buildings. This is because in many installations the excess heat over minimum comfort levels is wasted via additional transmission loss and also due to additional ventilation since doors and windows may be opened to compensate for overheating conditions in a given building.

Chilled Water Temperature Reset

A mechanical refrigeration system has a performance efficiency or coefficient of performance which varies with the difference between suction (evaporator) temperature and condenser temperature. To achieve lower chilled water temperatures in chilled water systems requires lower suction temperature and lower suction pressure. This requires more pumping work for the same output tonnage. In general there will be a difference in temperature between the chilled water output and the evaporator temperature of 5-10^{o}F due to finite heat transfer. Resetting the chilled water temperature upward by several degrees allows evaporator temperature to rise approximately the same amount. This reduces pumping work by about 2% per ^{o}F for the same cooling output. During the non-peak design point conditions of cooling when demand is less than capacity, it is practical to allow chilled water temperature to rise from a conventional level of 40^{o}F to as high as 60^{o}F and still achieve cooling. This can be done effectively through the monitoring of zone demands and setting the chiller temperature at the level which meets the demand of the zone of greatest need at the time. This will also eliminate cold drafts from occurring in other zones due to over-cooled air flows in the zone. The HVAC system will operate in most zones in a more steady mode and be subject to fewer start/stop cycles. The accrued or integrated savings that are possible depend on the variation from zone to zone that occurs

in cooling demand. If an average reset of only $3^{o}F$ upward is assumed, the savings will be about 6% in cooling energy use during the operating period of the season.

Multi Unit Operation Vs. Large Single Unit Operation

Chillers, boilers, furnaces, pumps and fans are generally available in a variety of sizes. In general also, the initial cost per unit output tends to reduce with size or capacity due to non-proportional cost avoidances in construction or installation. The option exists to satisfy a given application with a single unit or several smaller units operating simultaneously. Often at least two units will be installed for the purpose of redundancy in potential failure situations. The total capacity may be more or equal to the single unit capacity depending upon the failure mode analysis and criticality. By selecting multiple units to meet potentially variable demand and a fixed peak demand it is possible to take advantage of operating efficiency characteristics of the equipment. This concept may best be illustrated by an example. An air cooled water chiller may have accessory equipment energy demands such as condenser fans and circulator pumps of up to 20% of the chiller compressor demand. By operating at less than peak output, the accessory power becomes a larger porportion of the total usage. By shutting off the partially loaded units and dropping back to units operating at peak efficiency then energy savings are incurred. Each type of equipment must be reviewed in detail to arrive at the best situation for an application. In general, one must expect additional first installed cost due to the unit output

cost noted. However, in most applications, variable demand is the rule and a 10% to 20% decrease in operating energy can readily be achieved by load matching. In fans, output volume may be varied by parallel operation, provided the inter-connect is aerodynamically proper. Boilers and chillers may operate in parallel and with staged controls to drop unneeded units as demand varies. Logic is also available to allow uniform wear distribution so that the same unit is not always on the line or off the line in the same sequence of use. In heating systems, the efficiency effect of using staged boilers may not be as significant as for cooling and fan systems since the efficiency is less a function of load than for the latter. However, economy of operation through reduced accessories energy use can still be expected.

Optimization of Reheat Equipment Operation

The reheat system of HVAC incorporates a single source of air at a cool temperature adequate to meet demand of the zone requiring the greatest cooling and each other zone incorporates a heat input in the terminal unit to raise the duct air temperature to the requirement of the specific zone. The terminal unit may use induction or a duct heater with hydronic, steam, or electric input. In the future the Army buildings will only use reheat in specific applications and where "waste" heat is available for reheat purposes. This waste heat may be that from the air-conditioner condensers, which is discussed later. In existing buildings or those which are not conducive to renovation of the entire HVAC system, certain improvements in energy efficiency is possible by optimized choice

of the duct temperature to minimize reheat energy input. This saves in both the cooling requirement and in the heating requirement.

By incorporating control logic to reset the duct temperature to meet the requirement of the zone of peak cooling demand without reheat in that zone, the total reheat of all zones is reduced as well as the minimization of the cooling energy requirement. The savings in energy that can be achieved depends upon the locality and the variation experienced in the zones. Given known conditions, the savings in energy can be evaluated per year per CFM of air handling capacity per the example below.

$$E \text{ (Summer Savings) } \frac{BTU}{CFM} = \text{Cooling weeks/year} \times \text{operating hours/week} \times (4.5\ \Delta H + 1.08\ \Delta T_s)$$

$$E \text{ (Winter Savings) } \frac{BTU}{CFM} = \text{Heating weeks/year} \times \text{operating hours/week} \times (1.08\ \Delta T_w)$$

where ΔH is the summer cooling enthalpy reset on the duct flow and ΔT_s is the reset on summer reheat (^oF) and ΔT_w is the reset on winter reheat, typical reset ΔH is from 2 to 5 BTU/lb depending on the zone arrangement and the climate and use of the building. The summer dry bulb reset in duct air reheat can be from 3 to 8^oF and in winter from 5 to 10^oF. As an example consider $\Delta H = 3$, $\Delta T_1 = 4^oF$ and $\Delta T_2 = 7^oF$. This results in the following for a 26-week season for heat and cool each and 60 operating hours per week.

$$E \text{ (total) } \frac{BTU}{CFM} = 26 \times 60 \ \left[4.5 \times 3 + 1.08\ (11)\right]$$
$$= 39593 \text{ BTU/CFM/yr}$$

The detailed building analysis will determine the savings for another specific case. However, for existing reheat systems the optimized logic could materially reduce operating cost by perhaps $30/1,000 ft^2 per year by this approach over current non-optimized reheat operation.

The use of reject heat as a source of energy for reheat systems is another economy that Army construction is taking advantage of. This heat can be derived from several waste sources, but typically the condenser reject heat is used on a double bundle condenser system. The cooling compressor discharge gas contributes to heating the reheat source of energy, either by direct passage of the airflow over the vapor coils or the use of an intermediate fluid to carry off the heat to the point of use. If the cooling unit is considered to be essential to operation of the facility then the condenser heat is free of additional cost and displaces the requirement for other energy input.

The compressor gas discharge temperature depends upon the efficiency and design of the compressor and the conditions of operation such as suction and condenser pressure and refrigerant used. In some designs the compressor and motor are cooled by the gas or water as well. In others, the motor and compressor are air cooled. For refrigerant R-22, the discharge gas temperature will be approximately as follows for the operating parameters of a typical chiller which has an air cooled

condenser.

Compressor Vapor Discharge Temperature - R22

Chiller Water Temperature ^oF	Ambient Temperature ^oF 85	95	105	115
60	209	222	240	115
50	215	228	248	262
40	219	237	253	268
30	219	239	253	270
20	228	241	253	275

The gas discharges from the compressor as a superheated vapor and must be first cooled to saturation at the condenser pressure. For R-22, the amount or enthalpy of superheat varies with the load situation but for the above range of conditions ranges from 3837 BTU per ton cooling to 4298 BTU per ton cooling. This assumes a specific machine efficiency of operation and condenser pressure. The condenser saturation pressure will vary with condition but will have a saturation range from 95^oF to 125^oF for the above conditions. Therefore for a specific example noted, the amount of de-superheat available from the compressor gas discharge will yield a temperature spread from input/output of 209/95 to 275/125 in Fahrenheit temperature. This heat is readily usable in the reheat equipment. The latent heat of condensation is also usable, but difficult to absorb because the saturation temperature (95-125) is essentially the same as the reheat air temperature for typical applications and allows little approach temperature differential for heat transfer. In summary, about 1/3 of the cooling load is readily available to use as reheat energy with a theoretical availability of about 1.25 the cooling load. The latter

is difficult or impractical to achieve. In a 10 ton system, 1/3 of 120,000 BTUH is practical to recover or about 40,000 BTUH can be routed to reheat use. This is free of energy charge except for incidental energy to transport the gas and/or water through exchange equipment. As much as 150,000 BTUH is theoretically available from thermodynamic considerations. The cost of implementation depends upon the distances between the chiller location and the reheated zones for piping, etc.

Fan Volume Control and Reduced Pumping Power

The use of a variable volume output of the air handling system in Variable Air Volume (VAV) designed systems is an economy of operation with respect to fan input power/energy. The amount of energy saved depends upon the building use, or applications, and the details of the a) fan performance and b) variable output mode of operations. The simplest, but least, saving mode is to use a damper on the inlet or discharge of the fan but without turning vanes and with flow proportioning by area control.

From the fan curves, it is necessary to obtain the fan static pressure output at full rated (design) flow and at the reduced flow condition (s). The discharge damper type of control chokes off the flow volume. It is also necessary to know the estimated volume reduction duration in a typical week. Consider a simple case of a single stepdown from 100% flow to 75% flow as is typical of a VAV demand schedule. Assume 55 hours per week reduced flow rate. Also assume a typical fan characteristic although each will be different and the manufacturer's data

should be referenced. A backward inclined fan would typically have a 2-inch of water static head at 100% rated flow and 2.2 inches at 75% of rated flow. Neglect variation in efficiency and assume 70% at both conditions. The energy reduction is from the fan power equation:

$$E \text{ (fan energy savings/CFM/year)} = 52 \times 55 \times \frac{0.8\text{kw}}{\text{hp}}$$

$$\left[\frac{1.0 \times 2.0 - .75 \times 22}{6356 \times 0.7}\right]$$

$$= 0.18 \text{ kwhr/CFM}$$

This energy savings represents a 17.5% reduction in fan power for a 25% reduction in flow. The accrued saving is less than the flow proportion because of the ineffectiveness of the discharge damper mode of throttling flow. However, the savings is still appreciable and worthwhile to pursue for applications which are suited to the variable volume zone flow condition. If inlet vane control were to be used, the fan would perform with lower power than for discharge damper flow control. However, the introduction of the inlet vane device to control pre-swirl will introduce aerodynamic losses of up to 10% of the fan capacity in pressure rise while in the full flow condition. Therefore at full rated flow, the discharge damper may restrict flow by a 1-3% loss factor depending upon the velocity of the duct flow. The variable inlet vane device is more obtrusive and may reduce pressure use by 5%. No volume control will entail parasitic loss. Therefore in selecting a volume control, the potential penalty at full flow must be accommodated with the potential benefit at reduced flow. The key features in this evaluation are a) the flow demand schedule as a % vs. hrs/week and b) the fan and

control output characteristic vs. flow rate. Other methods of flow control such as variable speed and variable pitch do not affect full rated output performance and are of approximately equal effectiveness in energy savings but are both very expensive to implement compared with the previously mentioned types.

Consider a system which displays a 5% power penalty at full flow which occurs 113 hours per week. Using 20% power reduction at 75% flow output, yields a net annual average savings in fan power as follows.

$$\text{Net Annual} = .2 \times \frac{55}{168} - .05 = .015$$

Energy Savings = 1.5%

Therefore for the example system, the use of a variable volume control of the type noted would not materially reduce pumping energy when the loss of efficiency is included for the control element itself.

Fluid Pumps

Most HVAC systems incorporate water pumps to transfer heating and cooling media, condensate, cooling tower water, etc. Many of these pumps are electric driven and therefore consume high unit cost energy. In the past, the efficiency selection of a pump for a given service was secondary to price, delivery, or other non-energy related criteria. In the future, the efficiency of the pump-motor system will have to be evaluated as a primary consideration in selection. Consider first, the electric motor driven pump, directly coupled to the motor shaft. The electric motor efficiency is generally highest at near rated load. The motor selected for pumping should be adequate over the anticipated operating characteristic of the fluid system, i.e., head flow curve, and avoid any potential over-loading conditions. However, selecting a motor anticipated to operate at less than 3/4 of rated load under nominal conditions is inviting electrical inefficiency in operation as well as reduced power factor in electrical service. In selecting a pump, the mechanical efficiency at or near the expected operating point should be near peak efficiency of the unit. The added cost of higher efficiency pumps can be traded against improved lifetime energy savings. Often the choice is that of the size of intake and discharge pipe to characterize the pump. Also, the impeller size at a given speed can be selected for best operation. Careful selection of the pump and motor can mean the difference between operating at mechanical efficiency of 40% and at 70%. Recognizing that the pumping output must meet the requirement, this means that input energy for the above example may be 75% more for the

less efficient selection. If the pumped fluid is used for heating purposes, the dissipated energy assists in heating the throughput and is put to some use, although electric energy used for heat is an expensive option. If the pumped fluid is a coolant, the system load to chill the water is proportionately increased according to the dissipation of energy.

Variable speed operation of pumps to accommodate head-flow changes is not as prevalent as for air flow fans, although energy savings are possible. First, the amount of pumping power to transfer similar amounts of energy is different for water and air. Typical design flow velocity for air is 1,200 f.p.m. and for water 180 f.p.m. Head loss for these conditions are 0.1 inch water per 100 ft. for air and 30 inches of water per 100 ft. for water. However, the relative volumes of air and water are, for similar energy transfer, with air differential temperature of 40^{O}F and water differential of 20^{O}F as follows:

$$\rho_a C_{pa} V_a \Delta T_a = Q_{carried} = \rho C_p \; V_w \Delta T_w$$

$$\frac{V_a}{V_w} = \frac{\Delta T_w}{\Delta T_a} \; \frac{\rho_w}{\rho_a} \; \frac{C_{pw}}{C_{pa}}$$

$$= \frac{20}{40} \; \frac{62.4}{.075} \; \frac{1.0}{.241} = 1726$$

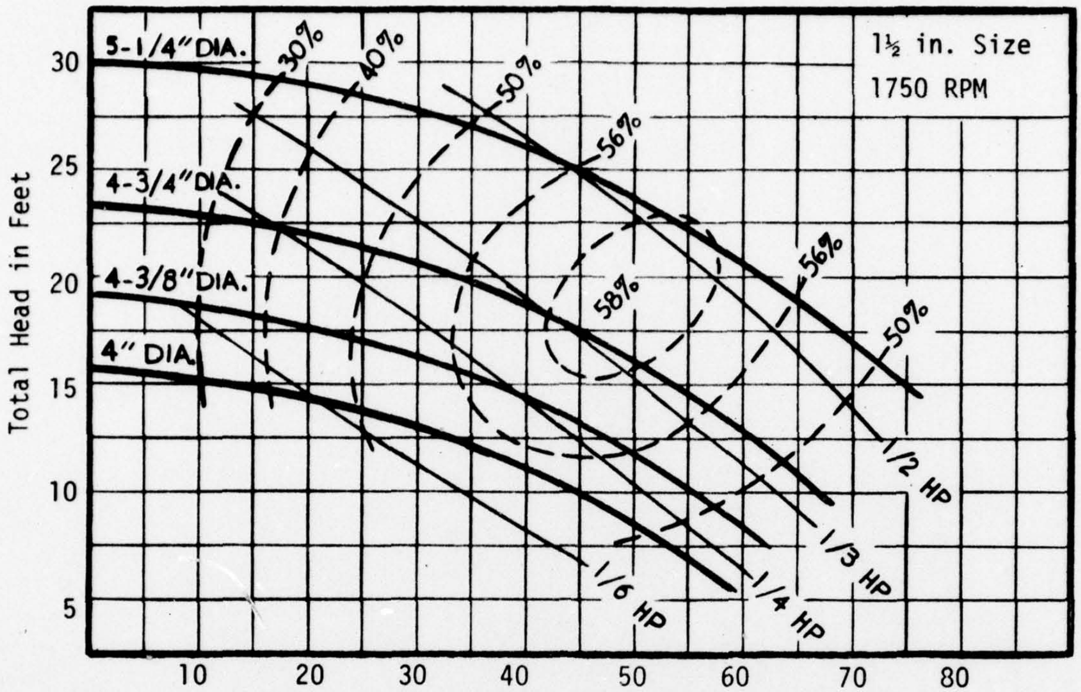
Therefore the energy input for typical design parameters, being the product of head and flow.

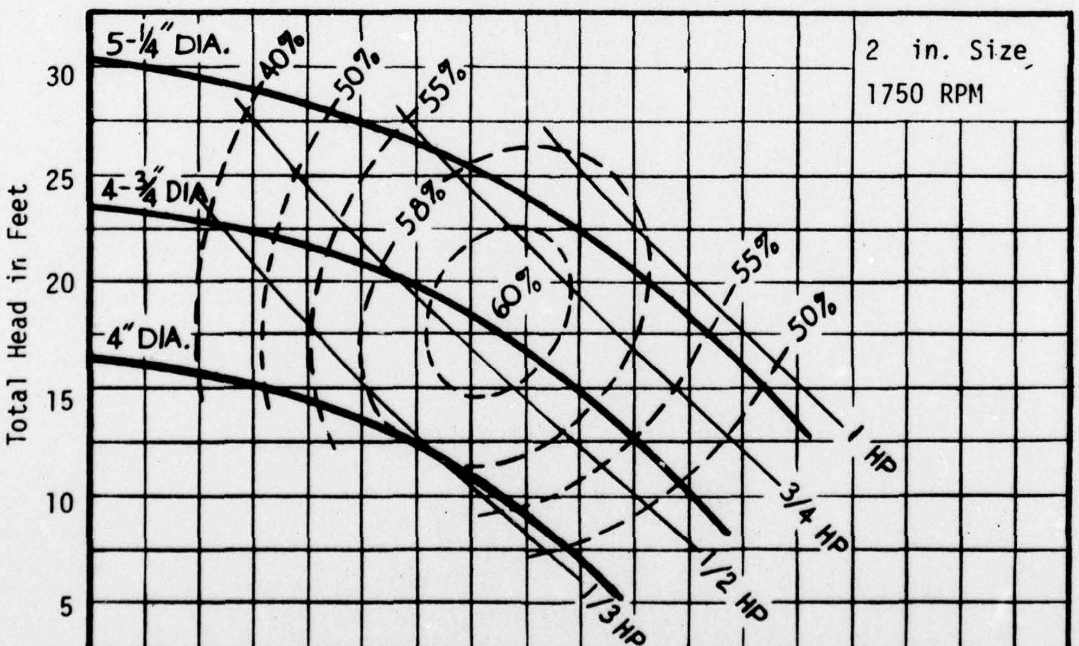
$$\frac{E_{pumping\ air}}{E_{pumping\ water}} = 1726 \; \frac{0.1 \text{ in water}}{30 \text{ in water}}$$

$$= 5.75$$

The result is that efficiency improvements in pumping air will reap large lifetime energy savings in conservation and cost whereas efficiency improvement in pumping water will yield less for the same initial investment in equipment. This means that high initial expense to achieve high water pump efficiency is less practical than the same expense for an air system. Improved hydraulics in the pump comparable to inlet vane control are probably not justified for conventionally sized pumps for HVAC systems. However proper initial pump selection for fixed speed, fixed geometry pumping is appropriate to explore improved efficiency versus cost. Generally, the selection of an optimum rotor size at the desired flow and head will yield highest mechanical operating efficiency.

A typical pump operating characteristic is presented in the attached figure for a hydraulic circulating pump of two different inlet/outlet sizes but with the same impellers and speed. The upper arm is for the 1 1/2 inch size and the lower for a 2 inch size. Anticipating a 40 GPM system flow and 20 ft. head requirement would indicate a pump operation efficiency of approximately 57% with the upper pump type (1 1/2) and approximately 53% with the lower curve pump type. Both would require a 1/2 H.P. motor drive. If instead the flow was selected to be 60 GPM at 20 ft. head, the upper pump would still develop 57% efficiency with an appropriate impeller size selection. The lower pump at 60 GPM and 20 ft. head would develop approximately 59% efficiency. Both would require 3/4 H.P. motors. Therefore, a change of 20 GPM in the same model pumps, speed, and motor size will allow a range of performance from 53% to 59% efficiency. At 60 GPM, selecting the lower pump over the upper will require less energy over the system lifetime just due to efficiency choice, i.e., 57/59 = .966 of the upper pump energy. For a motor of 0.92 electrical efficiency and water pumped





Typical Pump Performance Variation With Flow and Geometry.

at 60 GPM and 20 ft. head the power reduction is 15 watts of input. If the pump operates steady, as many circulators do, for a useful life of 10 years, this amounts to an energy savings of over 1,300 Kwhr and it is likely that no extra initial cost would be incurred in the purchase at installation since the basic pumps are the same. If the system was to run at 40 GPM instead of 60 GPM, such as would occur if the temperature differential were increased as has been discussed, then the upper pump at 57% efficiency would save 20 watts or 1733 Kwhr over the lower pump. Therefore in sizing and selecting pumps, the mechanical efficiency can provide energy savings at the initial selection. Choosing pumps to operate on or near flat efficiency characteristics will minimize the effects of inaccurate estimation of system head-flow characteristics upon the operating efficiency of the pump.

Improvements to lower water pumping energy are possible through the use of larger conduit pipe and higher temperature differentials such that the energy lost to friction compared to that transported is a small fraction and small in absolute terms as well. By increasing the working temperature differential on the water from a value of $20^{o}F$ as conventionally used at design point to $30^{o}F$, it reduces flow rates to 2/3 of the former. The contribution to pumping energy related to pipe friction will then reduce to approximately 45% of the previous value even at rated total energy transport. For chilled water systems the conventionally used 3 GPM per ton at 8^{o} rise can be extended to 12^{o} rise at 2 GPM at design point. At less than design conditions, the constant flow rate will result in lower rise of coolant temperature and permit good heat transfer performance. Two effects will tend to cancel any appreciable net influence on fan-coil or

duct-coil size related to the reduced flow rates and higher differential temperatures. These effects are as follows: a) the room temperature comfort standards have been changed in recent years to be higher in summer and lower in winter, and b) the changed room air temperatures will advantageously affect the heat exchanger surface log mean temperature difference but not significantly influence the film coefficient conductance.

The overall heat transfer conductance of a typical air-water heat exchanger is not linearly dependent upon the water through flow but rather to a dependence which varies with flow to the power approximately 0.8 and with a coefficient or influence factor less than unity. The result is that a 33% reduction in flow will influence the conductance by less than 10% in a typical application. The reduced flow will also influence the log-mean temperature difference so as to reduce the total heat transfer. However, by using larger differences entering and leaving, the influence can be negated. With lower room comfort settings in winter, discharge air temperatures from the unit can also be lower.

In summary, the pumping power input for transferring the heating/cooling media around the fluid circuit can be minimized through proper matching of the pump characteristic to the system characteristic so as to operate at or near peak efficiency overall. Also, larger design point fluid temperature drops through the heat transfer equipment will require lower flow rates to achieve the same energy exchanges at lower pumping power input.

Variable Air Volume HVAC System Critique

The relatively new concept of variable air volume systems provides a means of energy conservation and still maintains comfort conditions. The system, however, is not the answer for all building locations such as board rooms. In buildings having various activities a combination of constant volume and variable volume may be employed.

Existing as well as new "all air" systems are adaptable to VAV. The upper and lower limits of air flow and vapor pressure must be properly controlled to maintain comfort levels.

In the early days of variable air volume systems, a comparatively inexpensive method of control was variable inlet vanes on the supply fan controlled from a pressure tap with sensor located at some remote duct location with thermostatically controlled dampers at air terminals. While this method does maintain a constant static pressure at a single point in the system, pressures at outlets are always fluctuating. At higher pressures there is danger of drafts and insufficient ventilation and cooling at lower pressures. Balancing such a system is near impossible.

A system known as "bypass system" gets its name from the fact that by a motorized damper or other means only a portion of the air passes through the diffusers with the remainder being bypassed back to the air handling unit. This system is not a cost effective one and could result in control problems.

Another method of controlling CFM delivered to the conditioned space is VAV terminal boxes in conjunction with inlet vane control (or discharge damper on small units). The CFM delivered from such terminal units may

be controlled by self contained or room thermostats. The duct static pressure required to operate self contained units vary with manufacturer. This type system does offer an operating cost savings. However, control problems may arise when night set-back is employed.

A relatively inexpensive method of varying air volume in existing constant air volume single and dual duct applications is by use of factory set pressure independent valves in connection with pneumatic or electric operators thermostatically controlled. With the use of inlet vane control and building diversity considered, considerable savings can be realized.

A typical VAV system is designed for a flow capacity equal to the sum of the anticipated cumulative simultaneous demand for air flow (heat or cool). This sum is usually less than the sum of the peak demands since one or more zones can be anticipated to be in reduced demand due to a) inoccupancy, b) solar exposure variation, c) equipment use sequence, etc. Therefore a VAV system may have an air handling peak capacity of only about 80% of cumulative peak demand. In operation under steady continuous duty, this is adequate. However, the use of deep night set-back (up) for economy in energy use creates a problem in peaking capacity. A typical night set-back system will only anticipate to achieve a temperature about 5^{o} under (over) the set point at start of occupancy. For periods longer than overnight, say weekends in extreme outdoor conditions, the system capacity to respond to transients such as the set-backs when the thermal inertia of the building has been allowed to follow the setback is limited. In the A.M., all units may demand peak flows and the capacity to respond may not be available. Also, the techniques used to control fan output volume may use a point in the system to measure its parameter,

either static pressure or velocity pressure to arrive at an improper decision as to required output. A sensor located to measure static pressure at or near the duct run to the farthest zone from the fan will adjust output based upon this signal. If the flow to a few zones is high and a few zones low, then the average signal interpretation may not generate enough flow. Also, the flow drops to point of sensing will not reflect the integrated demand but rather a complex demand function not directly related to actual needed flow. The use of a velocity pressure sensor as opposed to a static pressure sensor is another option. Again, the location of the control sensing point will influence performance.

The logic and control mode used to adjust the VAV duct temperature is also a significant functional factor. In general, the capacity to heat or cool will be varied both by the terminal volume dampers and the duct air supply temperature. The choice of duct air temperature will influence the damper settings by the thermostat. The variable flow will accommodate some of the needed variation of capacity and the variable duct temperature by outdoor air reset or other will also contribute. The logic of control to adjust these two parameters and the point of sensing used vastly influences the functional performance.

Reset of Hot and Cold Deck Temperature of Dual Duct Systems Related to Velocity Pressure

In the dual duct system the cold and hot streams of air are mixed to meet demands of a specific zone in separate terminal equipment. As flow demand for cold or warm air varies, the ducts will reflect a raise or drop in static pressure at a reference point. If the flow demand for hot deck flow is reduced by the terminal dampers then pressure will rise as a result of lowered flow losses. This implies that demand is reduced and allows for a reduction reset in the hot deck temperature. Note that all zones should be similar in load demand or else the reset temperature will not maintain required mixed air temperature in the coldest zone. The reduced temperature will allow the other zones to achieve required mixed air temperature with less energy input. It is assumed that optional outdoor air reset of temperature is already in effect but due to variable load, all zones are not in equal demand or at peak demand. The savings in energy that are possible on a yearly basis are as follows for a dual duct system. The energy saved per CFM of air handling capacity are accrued in summer and in winter as a function of reset. Assume that 50% of the volume flow goes each to the hot and cold deck in normal operation.

$$E\ (\text{annual/CFM}) = 1.08 \times 0.5\, t \left[W_H\, (\Delta T_{HS}) + W_C\, (\Delta T_{HC}) \right] + 4.5 \times 0.5t\, W_C\, \Delta H_C$$

where ΔT_{HS} is the heating season hot deck reset ^{o}F and ΔT_{HC} is the cooling season hot deck reset ^{o}F and ΔH_C is the cooling season cold deck enthalpy reset. The typical values of the above are:

$\Delta T_{HS} = 5\text{-}10^oF$

$\Delta T_{HC} = 3\text{-}6^oF$

$\Delta H = 2\text{-}4$ BTU/lb

It should be remembered that this type of reset risks variation from the desired set point in some zones due to the reduced capacity to heat/cool. However for similar heat loads the extremes of the environment should not exceed acceptable range.

Economizer Cycles and Energy Savings

By monitoring outdoor air temperature and humidity (enthalpy) relative to the system return air temperature and zone demand, it is possible to mix outdoor air with return air to achieve "free" cooling. In effect, the same principle applies to opening windows when conditions warrant to achieve greater cooling from outdoor air than is practical from a limited ventilation system. Variable flow dampers linking outdoor air dampers and exhaust dampers allow for optimal blending. First consider outdoor dry bulb temperature monitoring and control for use of outdoor air for cooling supplement. The locality of the building has a considerable influence on the savings. It is necessary to know the DBT occurrence for the area on an hours per year basis. The following table gives the approximate % of the time that the cooling compressor can be off when the ambient air is below a pre-selected set-point allowing for meaningful cooling from outdoor air. The table is composed of data for 8 CONUS cities, as being typical.

City	Cooling Season % Off Time of Compressor	Setpoint Control Temperature ^{o}F
Atlanta	18	60
Chicago	27	60
Houston	17	60
Los Angeles	68	55
Minneapolis	29	60
New York	37	55
Seattle	64	55
St. Louis	24	60

The cooling load decrease is manifest from reduced cooling of ventilation air and also from reduced cooling to balance solar and transmission loads. The energy savings is computed as follows for a typical installation:

$$E\ (\text{annual}) = CFM \times V \times t \times 4.5\ \Delta H_C \times W_C$$
$$+ \left(\bar{T}_a - 78\right) \overline{UA}\ W_C \times 168 \quad \%\ \text{off-time of compressor})$$

In this function, $\bar{T}_a$ is the average cooling season outdoor temperature and $\overline{UA}$ is the bulk building average heat transmission per unit temperature difference indoor to outdoor. In a given building and locality the ventilation effect may dominate the transmission cooling effect. Also, the internal heat loads have been neglected except as may be included in the ΔH_C term above.

The DBT economizer cycle must be evaluated on the merits of each application. The implementation is relatively inexpensive and reliable. The choice is whether to upgrade to a complete outdoor enthalpy feedback on the selection of the operating set-point. The DBT economizer cycle is best

suited to the low relative humidity areas of the country with temperate outdoor temperatures and full sun situations. Cooling load is largely due to sensible heating and the outdoor air enthalpy is low enough that some latent cooling can occur as well. Another consideration is the occupancy factor. A DBT sensor system can be timed such that the free cooling can pre-cool the building with night air prior to daylight hour occupancy and therefore reduce daytime cooling start-up load after a night set-up in temperature. The relative advantage of the enthalpy type of economizer cycle may not yield enough extra savings to justify the added cost of the controls and logic. Caution must be exercised when combining several separate but inter-active energy saving modifications. Shutdown of vent fans during inoccupancy will generally save energy. However in the cooling season when the outdoor air temperature is proper, ventilation during inoccupancy is an attractive energy savings. The difference lies in the generalization and use of average properties and temperatures if logical controls can discriminate when venting is advantage-ous then improvement in energy use is possible in excess of that predicted based upon average temperatures for the season.

The enthalpy type of economizer cycle uses a comparison of outdoor air DBT and humidity (enthalpy) with those of the return air to adjust outdoor air intake above a minimum required for ventilation make-up of the occupants. Functionally this is analogous to the DBT economizer except for the comparison logic. The key to the energy savings in excess of those associated with the DBT approach is the amount of make-up air

normally required, the locality and weather conditions, and the building use. For an administrative building with 10% of the air handling capacity as make-up air, the following annual energy reduction was calculated for several CONUS cities for a 60-hour week operation (occupancy) and when compared with a DBT system set at various optimal changeover settings.

Annual Savings Through Use of Enthalpy Changeover Control
(Ton-hours Per Year Per CFM for 60 hrs./week Operation)

City	Compared to Economizer Changeover Setting 82°	77°	72°	67°
Birmingham, Ala.	1.988	.801	.241	.256
Flagstaff, Ariz.	0.0	0.0	.985	2.25
San Francisco, Calif.	.537	.317	.166	.691
Los Angeles, Calif.	1.185	.991	.787	1.255
Monterey, Calif.	.023	.0135	.092	.578
San Diego, Calif.	1.285	1.221	.976	.972
Colorado Springs, Colo.	0.0	0.0	.857	1.883
Washington, D. C.	1.987	.883	.327	.491
Homestead, Fla.	7.520	3.305	.754	.051
Chicago, Ill.	1.321	.569	.240	.257
Boston, Mass.	1.332	.595	.279	.336
Minneapolis, Minn.	1.029	.517	.162	.363
Billings, Mont.	0.0	0.0	.469	1.059
New York, N. Y.	1.811	1.023	.362	.247
Greensboro, N. C.	1.984	.898	.373	.347
Columbus, Ohio	1.423	.672	.258	.345
Dallas-Fort Worth, Texas	1.433	.602	.262	.428
Everett, Wash.	.127	.073	.043	.352

It should be noted that the locality has a large influence on the operational savings and that the trend varies from city to city as to the savings due to a rise or fall of the DBT base of comparison. This is because the annual occurrence of enthalpy and DBT of the cities average weather conditions is specific to each city. Note that in the Rocky Mountain areas, the enthalpy system accrues no net savings except at lower change over points. This accentuates the requirement that each situation be evaluated on its own merit with regard to enthalpy economizer or DBT economizer. For Army buildings with a summer set point of $78^{o}F$, the 67 to $72^{o}F$ change-over for DBT type of economizer is best. To obtain an intuitive feeling for the results of the table, note that the cost of cooling is about $\$2.00 \pm \$1.00$ per 10^6BTU or about 2.4 ± 1.2 cents per ton-hour of cooling water cooled condensers. For a building with a 20,000 CFM air handling capacity and 60 hours per week operation, the annual savings of an enthalpy system in an area that will yield 1 ton-hour/CFM reduction annually is, at current fuel cost, about $500 over and above the DBT system savings. The implementation cost over the DBT system is primarily in the added humidity sensor and logic.

Vapor Condensation Interior to Buildings in Cold Climates

In the Army buildings under the Corps' jurisdiction there are many locations in the CONUS and Alaska that winter conditions are so severe as to foster the condensation of interior humdidty on cold perimeter walls, windows, and doors. This propagates a potential health hazard for germ growth and leads to mold, mildew, and rust damage to surface coatings, frames and other metal parts on the outer perimeter of the building interior walls. This condition of inside wall condensation has been aggravated in recent years with a trend to tighter building air infiltration limits. Occupants and related latent heat generating functions of the building produce water vapor. In the past, with economical energy sources, the practice was to provide substantial ventilation through vents, exhaust fans, poor window and door sealing, etc. which permitted high air turn-over rates and considerable vapor transport. More recently the trend to low outdoor air exchange, tight construction, and use of non-combustion sources of heat, i.e. electric and heat-pump, eliminated furnace combustion air replacement and further economized on outside air entry. In prior years, the winter increase of interior relative humidity by humidifiers was almost universal practice. With the new and tighter construction, the building occupants and their functions, i.e. showers, cooking, smoking, etc. have generated as much or more humidity than is needed to meet comfort levels and standards in buildings in winter i.e. from 30% to 60 % R.H. This ventilation reduction often shows up as vapor

condensation on or in the walls of the building. The simplest and most obvious situation is where condensate builds up on the interior surface of an outside wall or window.

In the case of simple condensation on the interior wall, the wall surface temperature has been reduced to, or below, the prevailing dew point of the room air. Treating the room wall as a composite wall consisting of an overall thermal resistance R_w, with an outside air film coefficient of h_o and an interior air film coefficient of h_i, results in a predicted condition for incipient condensation on the interior wall or window of:

$$\frac{T_{DB} - T_{DB}}{T_{DB} - T_O} = \frac{1/h_i}{1/h_i + R + 1/h_o}$$

A typical outside wall h_o is about 6 for a wind velocity of 15 mph and usual properties for air and environmental factors. This value is not critical since the major resistance to heat flow is the wall, and to a lesser degree the interior air film. The naturally occurring interior film coefficient with forced convection and on normal room height vertical walls is approximately $h_i = 1.6$. For flat interior ceilings, the factor is approximately 1.5 in winter conditions. In calculations the 1.6 factor will be used. Therefore, the condition for interior wall condensation is:

$$\frac{T_{DB} - T_{DP}}{T_{DB} - T_O} = \frac{.63}{0.63 + R + .167}$$

The difference $T_{DB} - T_{DP}$ depends upon the relative humidity in the room air. The denominator $T_{DB} - T_0$ depends upon the room and the outdoor temperature. The wall resistance R depends upon the construction of the building. A typical older building would have R about 13 and in new construction R about 32 depending upon the site location of the building, the expected use and lifetime as predicting the most economical insulation effect. Also, at the windows, the old single glazed 0.125 inch glass can be compared with triple glazed 0.125 inch glass in a vertical position. The multi-glazed panels are with 0.25 inch gaps. For a site with outdoor design condition of -32^{o}F and indoor T_{DB} of 68^{o}F, this results in the following predicted onset of condensation. For the old building (R = 13), the condensation occurs on walls at any relative humidity above 82%. In the new construction (R = 32) condensation occurs above 90% R.H. On windows, condensation on triple glazed glass occurs above 40% R.H. and on single glaze at any R.H. above a few %. The areas where condensation and possible frosting will occur are in occupied barracks and recreation facilities where shower rooms and latrines see heavy service. In these areas, the presence of condensate (liquid or solid) will be difficult to keep off walls and impossible to keep off windows without additional operations. Another problem that is related is the night set-back or unoccupied set-back in buildings. Assuming a 10^{o}F set-back at night, with low air exchange will mean that the specific humidity will be the same, but the relative humidity will raise. As the wall mass thermally adjusts, the new interior wall temperature will drop and night condensation is possible. For example an R = 13 building and with 58^{o}F

at night will show condensate at a day-time condition of 68°F and 55% R.H. which is typical of an occupied space. Morning condensate on the walls is a good prospect in these buildings at low ventilation rates at night and daytime occupancy at or near 68°F and 50% R.H.

The reduction or elimination of interior wall surface condensation should be relatively simple to effect in many buildings. One approach is to assure that perimeter interior walls are not closely fitted with bookcases and other furniture which would reduce free connection and/or air currents from the HVAC system from "flow washing" the walls with room air. Also, this free access will assure that walls have a line of sight for radiant interchange with interior warm masses and sources of heat to increase the wall interior temperature. These effects will slightly increase the overall heat transfer transmission loss to the outside but only slightly. For example, the doubling of the interior film coefficient of the air side to h=3.2 will increase overall heat loss by about 2½% in the R=13 type building and less in the R=32 building type. Higher values of loss are noted if the single glazed windows are to be kept clear because very high interior conductances are necessary and the effect on losses is greater. The additional wall side connection or heat input is possible via use of radiant input at the baseboard to propagate a heat curtain at the wall or via use of air discharges at the boundary to stir the air and maintain a higher mixed air temperature near the walls.

In existing buildings with poor comparative wall insulation, the incorporation of ventilation air reduction via night shut-off of fans, better sealing and caulking, and elimination of miscellaneous vents may

propagate a condensate problem where none existed before. This will be aggravated by the winter night set-back of temperature and the overall practice of having the daytime temperature reduced from prior prevailing practice when the building was designed. The benefit in reduced vent air heat loss should offset any cost and energy lost to the enhancing of interior wall film heat transmission to avoid condensation. Single glaze window condensation is likely at most R.H. (above 5%) and with night set-back it is difficult to avoid. In most cases window frost build-up will occur until the resistance due to the ice layer increases the interior surface to the dew point.

A potentially more serious but unseen problem probably exists in many existing Army buildings due to condensation. With imperfect vapor barriers, the interior moisture may permeate the wall to a point where liquid condensation and freezing is possible. This dampness has a long term effect of causing deterioration of building materials, cracking of masonry due to freeze expansion, and other moisture problems. The short term (seasonal) problem is that of wetting the wall components and degrading their insulation properties, thus increasing heat transmission. The analysis to determine the point in the wall that liquid condensation and freezing occurs depends upon two characteristics of the wall, the thermal conductance and the vapor permeance, as well as indoor and outdoor wet and dry bulb temperatures. Therefore it is not feasible to generalize since the vapor diffusion will differ in rate from the thermal diffusion in different materials and with rather minor surface

treatments such as paint and plastic coatings or film. The presence of frost and/or water in the wall is at best a heat transmission problem and may, as mentioned, be a structural integrity problem. In severe climates the frost and water layer may thicken to affect much of the wall as the insulating quality deteriorates with moisture. Also, in any thaw cycle, the frozen areas will create liquid condensate which will perhaps flow to other parts of the structure via gravity. The only solution to the aggravation of this problem with the new low ventilation standards and set-backs is to use good quality interior vapor barriers of multi-layer lapped, taped plastic and/or epoxy based interior wall paint to reduce vapor mobility thru the wall. However in arctic areas where total thawing is not naturally achieved this retrofit may not solve the problem without the need to drive out accumulated moisture via localized wall heating at high levels.

In summary, the problem of moisture condensation in winter on interior wall surfaces and in structural outside walls will be more prevalent in the old buildings with the reduction of vent air and use of night set-back for low sensible heat ratio applications. Window condensation will become more prevalent as well. The solutions to interior wall condensation are relatively simple and effective. They can be achieved at low energy cost penalty. Condensation in the walls is a problem requiring detail analysis of each building and the possible requirement for improved vapor barrier at the inside surface.

EVALUATION OF ALTERNATIVES TO OUTDOOR AIR

Outdoor Air Requirement for Ventilation

Buildings of standard construction will naturally interchange air from the environment due to infiltration/exfiltration thru door and window cracks, roof vents and intentional mechanical ventilation. Conventional practice is to maintain slightly positive internal pressure and result in exfiltration. The concern for energy costs related to conditioning the ventilation air tends to minimize the amount of air make-up and encourage tight construction including door seals, weatherstrip, caulking, dampered vents and reduced purging air flows. The outdoor air serves two distinct purposes; a) to support occupants with breathable air i.e. proper oxygen and carbon dioxide levels, and b) to dilute and purge contaminants, particulates, odors and vapors.

The physiological requirement for breathing air by a sedentary or minimally active adult occupant is satisfied by a flow rate of about 1 CFM fresh air. Undesirable or unhealthy levels of carbon dioxide may develop for continuous exposure if flow is reduced below about 4 CFM per occupant depending upon the volume of air per occupant. Therefore, fresh air supply should meet these requirements for safe and healthy breathable air exchange. The building codes and standards almost universally require substantially larger flow rates per occupant. Fresh air intake less than 4 CFM per occupant would require a method of carbon dioxide absorbtion such as practiced in some closed environmental systems for submarines, spacecraft and diving apparatus.

There are sorbents with specific affinity to carbon dioxide, such as monoethanol amine, which desorb when steam heated and can be regenerated. However, the energy budget is usually such that the process is only justified in unusual environments as noted. In normal operation of a building ventilation system, these extraordinary carbon dioxide reduction measures are not practical. The ASHRAE Standard 62-73 specifies no less than 5 CFM of outside air per occupant to satisfy physiological requirements. The outside air, as found, may not be usable as ventilation air without pre-treatment, i.e. filtration, etc. The outside air in industrial or urban environments may contain: particulates, sulphur oxides, carbon monoxide, photochemical oxidant, nitrogen oxides and miscellaneous hydrocarbon compounds. These constituents result from automotive and stationary combustion emissions which are essentially universal in presence but highly variable in concentration. Therefore outside air will generally contain adequate oxygen and carbon-dioxide dilution capacity but also can contain objectionable contaminants as well. Therefore, the introduction of untreated outdoor air in large quantities does not, of itself, alleviate contaminant buildup in the controlled space. In the past, the ventilation rate has been substantially in excess of the 5 CFM minimum for contaminant dilution. This is because the energy waste was not an economic or policy burden as is the case today. Also, ambient air in days long past was probably cleaner in most instances than today and pre-treatment of outdoor air was perfunctory. Also, in the past the cost of filter

and air clean-up equipment exceeded the influence of the additional energy and fan capacity to achieve greater exchange and dilution. Recently the ASHRAE has published revised standards for ventilation outdoor air in the absence of treatment. These standards are reproduced in Table 6 for categories of building use and occupancy of common application. A minimum and a recommended ventilation rate is presented based upon occupancy and the absence of a specific building code specification. It is significant to note that the ASHRAE recommendation of Table 6 can be reduced to 33% of the stated value if particulate control to 60 mg/m^3 is achieved through filtration, and 15% of the stated value may be used if particulate and odor treatment is applied to the outdoor air. This does not allow reduction below the recommended minimum values for the application. The latter does not fall below 5 CFM per occupant as discussed previously. In the specification of future new and retrofit construction, most buildings will be satisfactorily operated with 5-10 CFM per occupant of outdoor air during the occupied period and essentially zero during the unoccupied period. The unoccupied period can be expanded to include the time approximately ½ hour after actual occupancy to ½ hour before departure. The building air contents and probable exchanges due to door actions on entry and exit of personnel will carry the ventilation requirement during these periods and are about an hour per day of ventilation energy losses. This savings may not be achievable in certain hazardous or high contaminant areas where clean air supply on occupancy must be assured. In general, toilet areas and other high

contaminant zones cannot allow recirculation of the air supply to them and therefore must meet the requirements of heating/cooling/ humidification as well as ventilation requirements for these areas. If the vents can be consolidated to permit heat recovery, then the loss from these 100% outdoor air situations can be reduced.

Reduction of ventilation use of outdoor air to about 5 CFM per occupant will require commensurate increases in the filter and clean-up requirement on the air before recycling back thru the supply system. The dilution of odors, etc. usually achieved by outdoor air must be accomplished by the actual removal of the contaminant from the air stream. The prior design technique was to dilute to the threshold of sensitivity or safety depending upon the contaminant involved. A later section of this study deals with the effectiveness, cost, and energy consumption of the equipment for clean-up relative to the energy, etc. for ventilation air. There will result a minimum life cycle cost where further clean-up equipment investment will not reduce the energy use or affect cost savings in future energy expenditure.

TABLE 6. VENTILATION REQUIREMENTS FOR OCCUPANTS

	Estimated persons/ 100 ft floor area	Required ventilation air, per human occupant			
		Minimum		Recommended	
		cfm	l/s	cfm	l/s
<u>Residential</u>					
Single Unit Dwellings					
General Living Areas, Bedrooms, Utility Rooms	5	5	2.5	7-10	3.5.5
Kitchens, Baths, Toilet Rooms	-	20	10	30-50	15-25
Multiple Unit Dwellings and Mobile Homes					
General Living Areas, Bedrooms, Utility Rooms	7	5	2.5	7-10	3.5.5
Kitchens, Baths, Toilet Rooms	-	20	10	30-50	15-25
Garages	-	1.5	7.5	2-3	10-15
<u>Commercial</u>					
Public Rest Rooms	100	15	7.5	20-25	10-12.5
General Requirements--Merchandising (Apply to all forms unless specially noted)					
Sales Floors (Basement and Ground Floors)	30	7	3.5	10-15	5-7.5
Sales Floor (Upper Floors)	20	7	3.5	10-15	5-7.5
Storage Areas (Serving Sales Areas and Storerooms)	5	-	2.5	7-10	3.5-5
Dressing Rooms	-	7	3.5	10-15	5-7.5
Malls and Arcades	40	7	3.5	10-15	5-7.5
Shipping and Receiving Areas	10	15	7.5	15-20	7.5-10
Warehouses	5	7	3.5	10-15	5-7.5
Elevators	-	7	3.5	10-15	5-7.5
Meat Processing Rooms	10	5	2.5	5	2.5
Pharmacists' Workrooms	10	20	10	25-30	12.5-15
Pet Shops	-	1.0	5	1.5-2	7.5-10
Florists	10	5	2.5	7-10	3.5-5
Greenhouses	1	5	2.5	7-10	3.5-5
Bank Vaults	-	5	2.5	5	2.5
Dining Rooms	70	10	5	15-20	7.5-10

TABLE 6. VENTILATION REQUIREMENTS FOR OCCUPANTS
(continued)

	Estimated persons/ 1000 ft floor area	Required ventilation air, per human occupant			
		Minimum		Recommended	
		cfm	l/s	cfm	l/s
Kitchens	20	30	15	35	17.5
Cafeterias, Short Order; Drive-ins, Seating Areas	100	30	15	35	17.5
Bars (Predominantly Stand-up)	150	30	15	40-50	20-25
Cocktail Lounges	100	30	15	35-40	17.5-20
Hotels, Motels, Resorts					
Bedrooms	5	7	3.5	10-15	5-7.5
Living Rooms (Suites)	20	10	5	15-20	7.5-10
Baths, Toilets (attached to Bedrooms)	-	20	10	30-50	15-25
Corridors	5	5	2.5	7-10	3.5-5
Lobbies	30	7	3.5	10-15	5-7.5
Conference Rooms (Small)	70	20	10	25-30	12.5-15
Assembly Rooms (Large)	140	15	7.5	20-25	10-12.5
Cottages (treat as single-unit dwellings) (See also Food Services, Industrial, Merchandising, Barber and Beauty Shops, Garages for associated Hotel/Motel Services)					
Dry Cleaners and Laundries					
Commercial	10	20	10	25-30	12.5-15
Storage/Pickup Areas	30	7	3.5	10-15	5-7.5
Coin-Operated	20	15	7.5	15-20	7.5-10
Barber, Beauty, and Health Services					
Beauty Shops (Hairdressers)	50	25	12.5	30-35	15-17.5
Reducing Salons (Exercise Rooms)	20	25	12.5	30-35	15-17.5
Sauna Baths and Steam Rooms	-	5	2.5	5	2.5
Barber Shops	25	7	3.5	10-15	5-7.5
Photo Studios					
Camera Rooms, Stages	10	5	2.5	7-10	3.5-5
Darkrooms	10	10	5	15-20	7.5-10

TABLE 6. VENTILATION REQUIREMENTS FOR OCCUPANTS (continued)

	Estimated persons/ 1000 ft floor area	Required ventilation air, per human occupant			
		Minimum		Recommended	
		cfm	l/s	cfm	l/s
Institutional					
Schools					
Classrooms	50	10	5	10-15	5-7.5
Multiple Use Rooms	70	10	5	10-15	5-7.5
Laboratories	30	10	5	10-15	5-7.5
Craft and Vocational Training Shops	30	10	5	10-15	5-7.5
Music, Rehearsal Rooms	70	10	5	15-20	7.5-10
Auditoriums	150	5	2.5	5-7.5	2.5-3.5
Gymnasiums	70	20	10	25-30	12.5-15
Libraries	20	7	3.5	10-12	5-6
Common Rooms, Lounges	70	10	5	10-15	5-7.5
Offices	10	7	3.5	10-15	5-7.5
Lavatories	100	15	7.5	20-25	10-12.5
Locker Rooms	20	30	15	40-50	20-25
Lunchrooms, Dining Halls	100	10	5	15-20	7.5-10
Corridors	50	15	7.5	20-25	10-12.5
Utility Rooms	3	5	2.5	7-10	3.5-5
Dormitory Bedrooms	20	7	3.5	10-15	5-7.5
Hospitals, Nursing and Convalescent Homes					
Foyers	50	20	10	25-30	12.5-15
Hallways	50	20	10	25-30	12.5-15
Single, Dual Bedrooms	15	10	5	15-20	7.5-10
Wards	20	10	5	15-20	7.5-10
Food Service Centers	20	35	17.5	35	17.5
Operating Rooms, Delivery Rooms	-	20	10	-	-
Amphitheatres	100	10	5	15-20	7.5-10
Physical Therapy Areas	20	15	7.5	20-25	10-12.5
Autopsy Rooms	10	30	15	40-50	20-25
Incinerator Service Areas	-	5	2.5	7-10	3.5-5
Ready Rooms, Recovery Rooms	-	15	7.5	-	-
(For Shops, Restaurants, Utility Rooms, Kitchens, Bathrooms, and other Service Items, see Hotels)					
Research Institutes					
Laboratories	50	15	7.5	20-25	10-12.5
Machine Shops	50	15	7.5	20-25	10-12.5
Darkroom, Spectroscopy Rooms	50	10	5	15-20	7.5-10

TABLE 6. VENTILATION REQUIREMENTS FOR OCCUPANTS
(concluded)

	Estimated persons/ 1000 ft floor area	Required ventilation air, per human occupant			
		Minimum		Recommended	
		cfm	l/s	cfm	l/s
Animal Rooms	20	40	20	45-50	22.5-25
Military and Naval Installations					
Barracks	20	7	3.5	10-15	5-7.5
Toilets/Washrooms	100	15	7.5	20-25	10-12.5
Shower Rooms	100	10	5	15-20	7.5-10
Drill Halls	70	15	7.5	20-25	10-12.5
Ready Rooms, MP Stations	40	7	3.5	10-15	5-7.5
Indoor Target Ranges	70	20	10	25-30	12.5-15
Museums					
Exhibit Halls	70	7	3.5	10-15	5-7.5
Workrooms	10	10	5	15-20	7.5-10
Warehouses	5	5	2.5	7-10	3.5-5
Correctional Facilities, Police and Fire Stations (see also Gymnasiums, Libraries, Industrial Areas)					
Cell Blocks	20	7	3.5	10-15	5-7.5
Eating Halls	70	15	7.5	20-25	10-12.5
Guard Stations	40	7	3.5	10-15	5-7.5
Veterinary Hospitals					
Kennels, Stalls, Operating Rooms	20	25	12.5	30-35	15-17.5
Reception Rooms	30	10	5	15-20	7.5-10

VENTILATION AIR CONTAMINANT LEVELS

The quality of air in buildings for Army applications must meet certain criteria for health, safety and comfort. In addition, certain special operations require particularly clean pure air to preclude damage to sensitive equipment. Consider first the requirements of human occupants as minimal criteria.

The fundamental cirteria are breathable oxygen and carbon dioxide buildup in the environment of the controlled space. The oxygen use requirement is quite low for sedentary individuals but increases as activity and metabolic rate. Prone and at rest the oxygen demand is about .01. At heavy work the rate is about .05. To estimate atmospheric air rate these may be multiplied by about 5 to get from .05 to 0.25 SCFM normal air. This does not require a very large ventilation rate per occupant. However, carbon dioxide buildup is a physiological toxin and must be avoided. Sustained concentration of 3% CO_2 must be avoided and concentration less than 0.5% CO_2 is more desirable. To achieve this the ventilation dilution of the generated CO_2 must be adequate. The CO_2 production is from 0.5 SCF/hr to 2.5 SCF/hr per occupant depending upon the activity level. These correspond to .008 SCFM and .045 SCFM of CO_2. Dilution to the desired 0.5% CO_2 requires from 1.6 CFM to 8 CFM per occupant for long duration exposure, i.e. steady occupancy. If the space is only intermittently occupied, then the CO_2 buildup will be slow and a transient will occur. This depends upon the activity level and the total volume of air per occupant in the space. In intermittent occupancy, with nightly vacancy, the transient condition may permit

lower ventilation rates. This depends upon the building use. However, the use of vent fan night shut-off for energy savings sake should be examined to assure that CO_2 buildup will not occur at full daily occupancy. As a general rule, in steady occupancy and average activity level, the CO_2 will not exceed the 0.5% threshold at vent rates in excess of 5 CFM or occupant. At this rate of ventilation, the oxygen level will be about 20.11% compared with about 20.62% in outdoor air and a minimal criteria limit of 17%. If a CO_2 absorber is used, then the vent air could be reduced but the nitrogen and other component buildup would require purging or absorbtion unless the vent rate was about 0.1 to 0.2 SCFM. Also, totally effective CO_2 absorbtion is not practical. The energy consumed in desorbing the CO_2 absorbtion bed or in cost of replacing the chemical prohibits the use of absorbtion equipment in all but unusual systems such as missile defense shelters, submarines, and space applications. Therefore the basic physiological need for long or steady occupancy is from 1.6 to 8 CFM with 5 CFM as an acceptable safe criterion average. The ASHRAE uses the latter value as a lower recommended limit in its standards. Note that in a building with short periods of peak use, such as chapels, etc., the low activity level and large volume per occupant will permit use of lower vent rates for the short term without incurring high CO_2 limit problems in the occupants. However, steady occupancy requires 3 CFM per occupant of vent air at minimal activity (sedentary) to achieve a 0.5% CO_2 limit. Therefore the range of from 5 CFM to 3 CFM carries some risk of CO_2 buildup depending upon the

occupancy period and activity level. Ventilation below 3 CFM per occupant in steady occupancy risks CO_2 buildup to hazardous levels.

The dilution of contaminants in the air generated in the space other than that noted above will require outdoor air or filtered return air. The acceptable quality criteria as noted in ASHRAE Standard 62-73 permit average levels of the following:

a)	suspended particulate	60 microgram/M^3
b)	sulphur oxides	80 "
c)	carbon monoxide	20,000 "
d)	photochemical oxidant	100 "
e)	hydrocarbons (excl. methane)	1,800 "
f)	nitrogen oxides	200 "

In addition, odor level should be non-objectionable and other toxic agents should be less than 10% of the threshold limit as established by OSHA or other standards agencies. In many cases the outdoor air cannot meet these standards without filter clean-up treatment. Naturally occurring air has an average carbon dioxide level of about 0.03%. In industrial areas and near large fossil fuel burning operations the levels may be substantially larger.

Particulate contaminants may be generated internal to the building as pick-up from the building construction materials or from processes performed by the occupants as part of this mission function or just casual occupancy. Smoking generates particulate (smoke) as well as odor constituents. Particulate may be inducted from outdoor sources of dust due to construction, excavation, agriculture, vehicular traffic,

etc. The role of the HVAC system is to provide air to the controlled space within accepted bounds of particulate levels. The ASHRAE Standard 62-73 permits reduction of outdoor air to 33% of suggested levels, not less than 5 CFM, if particulate control thru filtration is provided to levels less than 60 microgram per cubic meter. Filters are classified as to their efficiency to entrain atmospheric dust as a percentage. Medium efficiency filters trap 40-70% of particulate, better efficiency units trap 80-99%, and so called high efficiency particulate filters trap over 99%. Often the efficiency of the filter will vary with the particle size of the contaminant. In general, with dry media filter, the efficiency will directly correlate with the pressure drop of the air flowing thru the filter. Typical average face velocity thru these filters is in the range of 250 FPM to reduce velocity and to reduce pressure drop. The pumping power required for filtration of air is significant because: a) air filtration is usually continuous during ventilation periods and b) the entire vent airflow is usually passed thru the filters on each circulation. The outdoor air will also be rough filtered in most cases to reduce the particulate burden from outside. The air handling capacity in CFM which the filter passes is not a simple function of the space volume serviced. The capacity may be sized to meet the needs of: sensible heating or cooling, humidification, dehumidification or ventilation airchange requirements. Therefore the filter pumping power loss may be large or small depending upon the air

handling capacity. Typical filter initial pressure drops vary from 0.1 to 1.0 inch water column, and loaded pressure drops from .25 to 3.0 or in the ratio 2.5 or 3.0 to 1. During the useful filter life the average pressure drop will be about twice the initial loss. The fan work per 0.1 inch of water pressure drops thru the filter per CFM is as follows:

$$E \text{ (fan energy/CFM/year)} = \frac{52 \text{ weeks}}{\text{year}} \times \frac{168 \text{ hrs}}{\text{week}} \times \frac{0.1 \text{ inch}}{6356\ 0.7} \times \frac{0.8 \text{ kw}}{\text{hp}}$$

$$\frac{\text{Kw hr}}{\text{year}} \text{ per CFM} = .157 \text{ per } 0.1 \text{ inch of water}$$

Therefore, two design choices must be made, namely the air handling capacity thru the filter and the filter efficiency (pressure drop). Low efficiency will require higher air changes to achieve particle dilution, but will also incur lower pressure drop in general. A large dependence is whether the air handling capacity is set by ventilation requirements or by sensible or latent heat/cool requirements. For higher capacity systems, than required for ventilation, the filter bank may be of a lower efficiency and turn the air over at a greater frequency. The equilibrium particle density in the space is inversely proportional to the product of the volumetric flow and the filter efficiency. Therefore if the air handling capacity is set by the ventilation criterion flow, the choice exists to use higher efficiency filters at lowered vent flow or lower efficiency at higher vent flow. The ASHRAE Standard 62-73 acknowledges this by accepting 1/3 of the recommended vent flow of outdoor air if particulate control by filter to a criterion is achieved.

Each solid media filter has its own flow-pressure drop behavior. However, at face velocity of about 300 FPM the filter efficiency/pressure drop ratio is almost constant for efficiency to about 90% for atmospheric dust particles. Therefore the flow x efficiency product becomes a flow rate x pressure drop product or a filter work equivalent. The loss of energy due to filter drop for equal quality of space air is about the same whether the efficiency is high and the flow low or the efficiency is low and the flow is high. In examining first cost it is also found that the installed cost of high efficiency filters is higher than that of low efficiency filters but not in direct proportion. Therefore the cost effective choice will be to select higher efficiency filters that do not constitute a first cost or pressure drop greater than a linear function of efficiency. An energy loss and a cost handicap is incurred if the selection is made to have high efficiency filters on total flow if the flow capacity is established by criteria other than that for proper ventilation. For example, if the sensible heating of a space indicates a need for 10,000 CFM and the ventilation criteria based upon occupancy is only 5,000 CFM for the space, then using full flow filtration with high efficiency will incur an energy handicap of about 2x and probably over achieve in particulate control relative to the required level. However, the particulate contaminant density is directly proportional to the particle mass generation rate in the space. For high particle loads the density of particulate matter must be maintained below the criterion.

The first cost of low efficiency (40-75%) of filters is from $25-$40 per 1,000 CFM capacity and for medium efficiency (80-99%) filters from $30-$60 per 1,000 CFM. Therefore it is apparent that arbitrary selection of high quality filter operation <u>and</u> high volume flow rate will incur both energy and cost handicaps to the project, since fan work proportional to pressure drop and flow rate, and equilibrium particle density is inversely proportional to flow rate and efficiency. All of these factors must be evaluated to achieve acceptable filter conditions. The achievement of the 60 microgram per cubic meter or less particulate density will permit use of very low outdoor air rates, and therefore save on the energy required to heat/cool the vent air that would be required. Increasing the ventilation flow capacity just to increase turn-over in inefficient filters is not energy effective because the pumping power required to flow through ducts, grills, etc. is all attributed to the filter control function for that flow in excess of the sensible or latent heat satisfaction requirement.

In summary, the dry solid filter approach to particulate contaminant control in lieu of the use of clean outdoor air to dilute and purge the space can be energy effictive if properly designed. This requires that the air handling capacity be sized to the prevailing criterion, be it vent rate or heat/cool function. Then the filter efficiency and pressure drop comparative trade may be made. This requires the recognition that the filter operation creates a pressure drop generating a flow work input by the fan. If the interior particle generating load is high

and outside air is relatively clean then the ventilation system may use large amounts of outdoor air provided some energy recovery is achieved in the vent/inflow function.

There are numerous types of solid media filters as noted. The automatic electrically driven roll type replace the exposed filter media as it is filled. The replaceable pad type is prevalent and if changed frequently when needed, as noted by a pressure drop monitor, then the pumping power loss will not become excessive. The problem is usually that maintenance on the items is only infrequent. Washable filters save on material cost but increase labor cost. Also, the washable pads and screens are not as effective as the viscous treated fiber type of filter. The fabric type of bag filter is more effective at small particle size but more expensive.

Another type of filter used to reduce airborne particulates in process areas is the cyclonic types. These are useful to reduce relatively high loads of larger particles over 1 micro meter in size. These use a vortex principle to centrifuge air and separate the particle load. Their application is, as mentioned, usually limited to process requirements for industrial and agricultural dust problems. All of the mechanical type separators which use vortex action or a fan type of centrifuge consume substantial energy because of the high gas velocity developed and the high shear rates induced to achieve the separation of solid and gas. Their application should be carefully evaluated against other options because of this.

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STUDY OF ARMY BUILDING VENTILATION SYSTEMS FOR ENERGY CONSERVAT--ETC(U)
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MICROCOPY RESOLUTION TEST CHART

NATIONAL BUREAU OF STANDARDS-1963-A

There are two active types of particulate filter systems in prevalent use in HVAC systems. The active feature indicates a power input is required to accomplish the function. These two types are a) the liquid scrubber (air-washer) and b) the electrostatic precipitator. The air washer was prevalent in older systems and the precipitation in newer construction.

The air-washer uses a dense spray of an aqeous solution, often just water, to precipitate particles and soluble vapors into a reservoir basin. The units usually include a particular fog eliminator section where impingement of the flow eliminates the majority of the water droplets from the air. This type of filter by its very nature influences the humidity of the air stream. Therefore it is used to control humidity as well as filter particulate soluble vapor, and if containing air oxidant it will control other vapors common to HVAC contaminants. The air washer may use a chilled water in summer to achieve dehumidification and a warm water in winter to achieve humidification where low relative humdity is a problem. However, the energy going into/out of the latent heat portion of the process must be accounted for in the energy balance. The use of ambient temperature water to reduce evaporation and either a liquid filter or continuous flush of the residue basin of the filter will minimize the energy input. The cost of the water and/or chemical additive must be included in the operating cost of air washers since it is usual to flush the residue tank to assure pure water flow. This type of filter requires more power input than typical solid filters since it contains a fluid pump and some flow pressure loss as well as the latent heat problem noted. In the case of chemical additives for odor control the

chemical feed cost must be added. Chemicals may be of the once through or the regenerating type. In either case the regenerating cost or disposal cost must be included in the operating budget. The advantage of the liquid scrubber system is that it can remove particulate from about 0.2 micron up with very high efficiency and the pressure drop does not increase with loading as with some types of fixed pad solid filters. Liquid droplet entrainment for high air through flow velocity is a problem in this type of filter. Also, the maintenance of a wet system has all of the attendant problems of corrosion, leakage, pump seal failures, etc. It is usually necessary to add reheat to the output of the scrubber air stream since the latent cooling occurs. The air washer filter system has advantages in applications where its humidification/dehumidification property is of HVAC advantage such as in recreational facilities. Also, the ability to remove vapors and odors is an obvious advantage in many applications. However, this topic will be discussed in the next section. The energy input associated with the (de)humidification effect may or may not be attributed to the scrubber depending upon the sensible heat to total heat ratio of the controlled space. The first cost is rather high for these systems and with low cost energy, the practice was to use solid media filters and high air flow to achieve particulate control while using what are now deemed very high outdoor air ventilation rates to dilute particulate density. The increased cost of energy, reduced ventilation flows and general degradation of outdoor air quality in many regions of the CONUS have re-kindled interest in the air washer. Also,

the ability to achieve vapor control is important in sophisticated Army missions where chemical processes, vapor degreasing, plating and other hot process operations take place which generate fumes and vapors. The air washer type filter can remove water soluble vapors and gases very effectively. In swimming pool areas where chlorine compounds are prevalent and air turn-over requirements are high to dilute the vapor and carry off the moisture of evaporated pool water, the air washer using ambient source water can reduce humidity without mechanical or electrical power input of any great magnitude. This also permits the reduction of outside air and the attendant energy loss associated with air exchange. By flushing the washer sump with fresh water, the dissolved compounds can be removed to a disposal point where they are harmlessly passivated to an acceptable byproduct.

The other category of filter for particulate is the electrostatic precipitator. In this type of unit, which operates dry, the dust particles are electrically charged and electrical fields deflect particles to a capture area for shakedown and disposal to a sump. This type of filter is effective to remove airborne particulate in the range from .01 to 10 micron sizes. It is efficient enough to remove most of the common contaminants of HVAC interest. The high first cost and maintenance cost limits its application to those applications such as health care facilities where particle control is medically important.

Odor Removal

The reduction of offensive odors present as vapors or gases in the air is a difficult problem for any HVAC filter system. The reason is that the concentration for the threshold of objectionable sensation is low, often much less than 1 part per million. The presence of odors can cause a sense of poor environmental control even when temperature, humidity, and basic ventilation comfort conditions are met. In the rehabilitated and new construction the lowered vent rates are such that the dilution of odors will be less effective. The ASHRAE has developed recommendations for odor free air supply for occupant generated odors. This recommendation is presented in the following Table 7. Note that the dilution required varies with the air volume per occupant and the other prevailing air conditions of temperature and humidity. Note also that the air supply of odor free air exceeds the minimal rates required for physiological conditions of oxygen and carbon dioxide limits. The use of an air washing dehumidifier in summer shows that for sedentary adults, the air supply can be reduced to less than 4 CFM per occupant if occupant volumetric density is about 200 cu. ft. per person and the total air circulation is about 30 CFM per person. The air space per person is significant because the odor constituents generated by the occupant dilute into the air space and are subsequently flushed by the fresh air supply and the circulation currents in the space. The outdoor air requirement is about 5 CFM per occupant,

or slightly less, but the odor free fresh air supply for comfort is from 1 to 5 times this for adults in Army type applications. However, this odor free air need not be outdoor air, but may be recirculated, scrubbed air. The efficiency of the wet scrubbers can be noted in the attached Table 8 . In summer the odor free air is less than 4 CFM per occupant or about that required for the physiological needs. This is significant because the outdoor or make-up air can be reduced, and outdoor air heat/cooling losses are minimized by using recirculated but cleaned air supply. There will be an economic trade-off of added energy to use higher outdoor air rates and less expensive filtration/clean-up or to use lower outdoor air rate and comprehensive filtration and clean up.

Through the use of face and bypass dampers on the filter/clean-up system it is possible to route only a portion of the circulation air thru the cleaner and the remainder passing directly through the supply system to the particulate filter and/or the air handler in the air system. As a general rule, the use of extra outdoor air is to be avoided because of energy penalties. The odor removal requirement may be the pacing item in a determination of the outdoor air selection relative to better filter and clean-up. The air circulation rate may be determined by the heating requirement, the cooling requirement, the humidity requirement, or the ventilation and filter-clean-up requirement for particulate, metabolic gases, and odor. If the air flow circulation is 15 to 30 CFM per occupant during occupancy, as is typical of the heating/cooling requirement, then the metabolic requirement would have outdoor air (5 CFM per occupant) at 15 to 25% of the air handling capacity. The

TABLE 7. MINIMUM ODOR-FREE AIR REQUIREMENTS TO REMOVE OBJECTIONABLE BODY ODORS UNDER LABORATORY CONDITIONS

Type of Occupants	Air Space per Person, Cu Ft	Odor-fee Air Supply, CFM per Person
Heating season with or without recirculation. Air not conditioned.		
Sedentary adults of average socio-economic status................	100 200 300 500	25 16 12 7
Laborers.........................	200	23
Grade school children of average socio-economic status...........	100 200 300 500	29 21 17 11
Grade school children of lower socio-economic status...........	200	38
Children attending private grade schools........................	100	22
Heating season. Air humidified by means of centrifugal humidifier. Water atomization rate 8 to 10 gph. Total air circulation 30 cfm per person.		
Sedentary adults.................	200	12
Summer season. Air cooled and dehumidified by means of a spray dehumidifier. Spray water changed daily. Total air circulation 30 cfm per person.		
Sedentary adults.................	200	4

TABLE 8. WET SCRUBBERS AVERAGE COSTS 1976 PRICE LEVEL

Saturated Gas Flow Leaving Scrubber CFM	Manhours To Erect In-Place*	Dollars Per Saturated CFM Medium Efficiency	High Efficiency
500	20	$ 2.40	$ 3.20
1,000	28	2.24	2.96
2,000	34	2.13	2.72
3,000	38	2.05	2.56
4,000	42	2.02	2.45
5,000	46	1.98	2.40
6,000	50	1.95	2.32
7,000	55	1.92	2.27
8,000	60	1.89	2.24
9,000	65	1.86	2.22
10,000	70	1.82	2.18
12,000	80	1.80	2.14
15,000	90	1.79	2.11
20,000	100	1.78	2.08
25,000	120	1.77	2.05
50,000	150	1.75	2.00
100,000	180	1.75	1.98

Included: Heavy duty stainless steel scrubber, gaskets and start-up instructions.

Not included: Foundation, erection, induced-draft blower, pump, insulation, ductwork, control panel, wiring, piping, emission testing or special accessories.

* Special rigging not included.

75 to 85% of circulation air would recirculate through filters but not be vented outdoors. As was mentioned previously, the fraction of air passed through particle filters depends in large part upon the filter efficiency.

The odor removal methods are to use a dry method containing a packed bed of granular material, or a wet method using a spray type of scrubber. The dry methods may use a non-polar or a polar media to remove chemical vapors (odor). The polar types include alumina beds. The non polar types include activated charcoal (carbon) from a variety of sources to achieve various areas per pound of filter material. The charcoal types absorb most hydrocarbon type of odorants and are simple and inexpensive to reactivate. The charcoal beds are usually removed or recycled to drive off contaminants and to be re-activated. The alumina can also be reactivated by heat and/or steam. The average installed cost and annual operating cost for material is presented in the following Table 9 for activated charcoal filters in various typical applications. From this data, it is possible to evaluate the relative merits of the use of a charcoal filter in lieu of outdoor air energy make-up. The use of coated filter media which specifically reacts with given odor vapors is becoming more prevalent in new applications. This type of filter actually chemically reacts the odorant to a harmless or non-odorant gas with some products of reaction remaining in the filter bed itself. The specificity of the filter additive must be matched to

the odor source chemical. Therefore it is difficult to generalize on price or cost of reactivation. The following Table 10 provides the cost estimate for wet type of scrubber systems. The cost of an installed wet scrubber for various through flow capacities is presented for plain water type of washers. The chemical type and its associated special pumps, tanks, valves for chemical additive is not included in the price. It must be recognized that the output air is saturated at the liquid temperature saturation value, and therefore any energy absorbed to humidify or the associated reheat must be allocated to the washer unless humidification is necessary for other reasons.

Odorant removal by active chemical means is also possible. Potassium permanganate in aqueous solution or other metal ion salts can be used with air scrubber equipment to actually react the odor causing material to passive products. Another chemical means which is becoming increasingly applied, particularly in medical and health service facilities is the ozone generator operating from an electrical discharge. The ozone rapidly oxidizes odors and also is an air sterilizing agent which reduces pathogenic populations. This energy consuming, active approach has several drawbacks to general application. The ozone generated is itself a hazardous, toxic, gas and will affect the occupants and/or space contents if allowed to increase in concentration above the ppm level. This highly reactive oxidant must be monitored to assure that dangerous levels are not reached.

TABLE 9. ACTIVATED CHARCOAL FILTERS 1977 PRICES

Average Odor Index

Application	Odor Index	Cubic Feet of Space Treated by One Lb. of Charcoal
Laboratories	X-Heavy	50-400
Hospitals	Heavy	100-400
Offices, Private	Heavy	100-500
Bars, Taverns	Heavy	100-500
Theaters	Medium	300-900
Restaurants	Medium	300-900
Offices, General	Light	500-900
Department Stores	Light	500-900
Hotels	Light	500-1000
Public Buildings	Light	500-1000
Residences	Faint	1000-3000
Churches	Faint	1000-3000

Average Costs of Air Purification per Odor Index

Odor Index	Lbs. of Charcoal Per 1000 CFM	Cost per 1000 CFM of Purified Air	Average Cost Per Year For Reactivation
Faint	36	$ 365.00	$ 84.00
Light	44	445.00	100.00
Medium	60	620.00	135.00
Heavy	66	685.00	150.00
X-Heavy	72	740.00	165.00

TABLE 10. ELECTROSTATIC PRECIPITATORS
High Voltage

Average Purchase Costs - 1977 Prices

Gas or Air Volume Through Collector ACFM	---- Dollars per ACFM ---- Medium Efficiency	High Efficiency
10,000	$ 3.00	$ 4.00
30,000	2.90	3.85
60,000	2.70	3.70
80,000	2.60	3.60
100,000	2.50	3.50
200,000	2.20	3.00
300,000	2.00	2.60
400,000	1.80	2.30
500,000	1.65	2.20

For temperatures up to 650°F.

Included: Precipitator, high voltage assembly, lagging, control panel, key interlocks, hopper, heaters, and start-up instructions.

Not included: Foundation, platforms, ductwork, erection, inlet diffuser, piping, wiring, induced-draft blower, dust hopper valve or testing.

NOTE: Low - voltage equipment usually costs 50% - 65% less than high-voltage equipment. Use 0.6 multiplier.

VENTILATION ENERGY LOSS MODELLING ANALYSIS

Ventilation Air Energy Balance

Outdoor air introduced into the building controlled space must be conditioned by the environmental control system. The energy required for this becomes part of the heat/cool load for system capacity sizing and energy consumption. The air exhausted from the space will be at the control condition, since it will be from the mixed return air duct or direct vent from the space. The exception is when an exhaust energy recovery system is installed. The air entering from the outdoor air supply duct will in general exceed the air vented from the return air duct since venting or exfiltration will occur at toilet vents, doors, window cracks, etc. Therefore in energy analysis the relative flow rates and sources must be discriminated.

The calculation of the energy associated with ventilation losses requires certain thermodynamic analysis and concept developments. The space air handling capacity in CFM is a paramount consideration. This is the flow through the air supply system and includes the make-up outdoor air and the return air. The next definition is the fraction or percent outdoor air make-up (fresh ventilation air). This is easiest to express as a decimal (V) and is the ratio of the ventilation air to the total air flow through the air handling system. For existing buildings this may be from 0.1 to near 1.0 depending upon the application. The next concept is the energy loss associated with each CFM of outdoor air.

This is from fundamental principles, based upon sensible energy, given by the relation for air

$$Q\ (\text{BTU/CFM}) = 1.08\ (\text{average indoor temperature - heating season average outdoor temperature}) = 1.08\ (\overline{T}_i - \overline{T}_a)$$

For the cooling season, the enthalpy difference is more appropriate since latent heat is a consideration in summer cooling.

$$Q\ (\text{BTU/CFM}) = 4.5\ (\text{average cooling season enthalpy difference outdoor-indoor}) = 4.5\ \Delta H_c$$

The energy penalty is best evaluated on an annual basis for the full heating season and for the cooling season combined. Also, the hours occupied vs. unoccupied must be evaluated and included in the calculations.

For each geographical location, there will be a typical average heating season in weeks per year (W_H) and an average cooling season weeks per year (W_C). For this location there is also available an average outdoor season temperature (heating season) or enthalpy load (cooling season).

The building occupancy history will determine the average hours per week of occupancy. It is assumed that the ventilation system only is operated during the occupied hours. If these hours are different, the accommodation is required in the use of the energy formula. The hours per week of ventilation system use is (t).

The total annual energy use for heating/cooling the ventilation outdoor air is:

$$E \text{ (annual/CFM airflow)} = Vt \left[W_H \times 1.08 \, (\overline{T}_i - \overline{T}_o) + W_c \times 4.5 \, \Delta H_c \right]$$

For the Washington, D. C. area with an average heating season outdoor temperature of 44^{o}F, heating season of 28 weeks, a cooling season of 24 weeks and average cooling season outdoor temperature of 77^{o}F (Δ H = 6.4 BTU/lb), the above becomes:

$$E \text{ (annual/CFM airflow)} = Vt \times 1417$$

For a 25% (V = 0.25) make-up air system and full-time, 168 hours per week, ventilation this amounts to an energy usage of 59,514 BTU/year/CFM air handling capacity. For a modest sized system of 10,000 CFM the annual energy waste is 595 million BTU for the Washington, D.C. example. Analogous results may be obtained for other locations per Table 5 containing geographical data estimates.

In the above example calculation, the heating season effect on humidification or latent heat energy was neglected. With winter outdoor temperature average being low in general, and new D.O.D. guidelines on comfort allowing reduced humidity levels in administrative and occupied space in winter to as low as 30% R.H. @ 68^{o}F, the influence of latent heat is minimal. For example, for a case of outdoor air at 44^{o}F and 60% R.H. to raise to 68^{o}F and 30% R.H. the enthalpy change is 7 BTU per pound and the example calculation would provide 5.8 BTU per pound. The difference represents the latent heat which may or may not be supplied by the heating system depending upon the nature of the occupancy and use

of the building. The example will, in general, provide a conservative or underestimate of the energy loss incurred in ventilation air which is readily calculated from available data.

The cost of this energy loss, exclusive of the pumping power may be computed from the energy source cost. This is available from facility records or utility suppliers. Future rate inflation should be used for new projects occupying at future dates. Some examples are provided for sake of estimation:

Source	Unit Rate	Efficiency	Heat Energy Cost ($\$10^6$B)
No. 2 Oil (129,000 B/gal)	$ .45/gal.	70%	$ 4.60
District Steam (1,000 B/lb)	$4/Mlb	100%	$ 4.00
Propane (91,500 B/gal)	$ .35/gal	70%	$ 7.03
Nat. Gas (1,050 B/CFT)	$1.75/MCF	70%	$ 2.50
Elec. Resistance	$ .02/kwhr	100%	$ 5.80
Heat Pump	$ .02/kwhr	3.8 C.O.P.	$ 1.50

The above data is provided as being representative only and does not imply an actual ranking of costs since in any locality, one or another of the energy sources may be more expensive than the other due to relative geographical factors on supply, etc. The heat pump in particular is subject to geographical effects both due to electricity cost and due to outdoor temperature effects on performance (C.O.P.). Many heat pump applications require use of auxiliary heat below a pre-selected outdoor ambient temperature. Therefore the low energy cost of the heat pump is misleading if these other factors are not considered.

The cooling energy cost is that associated with the operation of the vapor compression air-conditioning system. Typical operating efficiencies of systems are provided below for the two prevalent cooling systems, i.e. air-cooled condensers and water-cooled condensers for electric motor driven equipment. These average values do not discriminate any efficiency effect with size but just that usually associated with the cooling rejection method. Also, the cost per million BTU cooling depends upon the load relative to design capacity load. However the following averages are representative barring other more specific data. When making a specific equipment selection, the actual operating performance should be referred to for relative energy use.

	Cooling Cost (Typical)	
Electricity Cost $/kwhr	*Water Cooled* Condenser $\$/10^6$ BTU	*Air Cooled* Condenser $\$/10^6$ BTU
.02	$1.66	$2.00
.025	$2.08	$2.50
.03	$2.50	$3.00
.04	$3.73	$4.00

The total system energy cost of conditioning ventilation air is the sum of the heating season cost and the cooling season cost. In some cases the time of operation per week or the ventilation rate may differ summer and winter due to variable use rates in seasonal training facilities for example. This can be accommodated by using different values of Vt for the heating and cooling periods.

The additional pumping energy to transport the ventilation air into the building can also be calculated although it should be recognized that the air-handling system may circulate the full capacity independent of the make-up air. For buildings with separate ventilation fans, the energy of pumping is given by the following:

$$\text{Kwhr used} = t \times \frac{\text{CFM} \times \text{Static } \Delta P \text{ (in water)}}{6356 \times \eta} \times \frac{0.8\text{kw}}{\text{h.p.}}$$

The efficiency is the fan mechanical/electrical overall efficiency and t is the operating hours per year. For example a continuous duty fan of 1000 CFM capacity, 2 inch of water ΔP and 70% efficiency will use 3150 Kwhr per year in ventilation air energy consumption. The reduction of operating hours will save both the ventilation energy of the prior section but also the vent fan energy as calculated in the immediately preceeding analysis if a separate vent fan is used to draw outdoor air into the building. If fans are used on supply of outdoor air and also on discharge then the above savings will be larger for each hour of shut-down, and may be estimated from the specific pressures, etc. The specific HVAC duct system selected for each application will dictate the pressure loss energy associated with fan work.

It should be noted that in the heating season, enclosed fan drives, i.e. where the motor is close coupled to the fan and in the duct air stream, the motor work contributes to the systems heating capacity. If the motor is outside the duct, only the pressure work contributes to the heating. In summer cooling the motor work adds to cooling load.

Pressure work resultant from high pressure or high velocity duct system selection should be compared as to energy cost and consumption for pumping.

The introduction of outdoor air based upon building area, which was a common design criterion in the past is wasteful of energy if the building is not occupied. A typical past criterion was 0.2 CFM/sq. ft. continuous ventilation for public buildings in normal use. A 25,000 sq. ft. facility would ventilate with at least 5,000 CFM not including any special needs of toilet rooms or other areas. If the 25,000 sq. ft. building is an office occupied on the average of 60 hours per week by 250 people (1 occupant per 100 sq. ft.) then minimum demand for outside air might be 5-10 CFM per occupant during occupancy. It is interesting to compare the relative ventilation air annual energy demand for these cases.

$$\frac{E\text{ (annual) } \begin{array}{l}0.2\text{ CFM/Sq. Ft.}\\ \text{Continuous}\end{array}}{E\text{ (annual) } \begin{array}{l}5\text{ CFM/occupant}\\ 55\text{ hours/week}\end{array}} = \frac{5{,}000}{1{,}250} \times \frac{168}{55} = 12.2$$

In this case ventilation is assumed to be off the first and last ½ hour on 5 days per week during occupancy.

The use of full time ventilation at excess rate is illustrated to be 12 times the energy demand of accepted minimum levels of ventilation when only used during occupancy periods typical of administrative areas.

ENERGY RECOVERY METHODS AND EVALUATION OF OPTIONS

Ventilation Air Energy Recovery System Analysis

The option exists in new construction and in certain rehabilitation projects to use an interchange of energy of the exhaust air to the make-up air to pre-condition the ventilation air flow. This is because the exhaust air is at or near the space conditions in the controlled zone while the incoming make-up air is at outdoor conditions. In general, the exhaust and supply air flows will not be exactly equal. This is because certain airflows exfiltrate at doors, cracks and vents or at exhaust locations far from the entering air location. The outdoor supply will in general be the larger of the flow quantities. Due to finite efficiency effects, the outgoing and incoming air cannot totally interchange states even if airflows were equal. Also, the interchange may be limited to heat flow as in sensible heat exchangers, or it may include mass exchange, i.e. water exchange as in the enthalpy type regenerative exchangers. The type of recovery i.e. sensible heat only or enthalpy and the sophistication of the exchange system depends upon the application, the location geographically and the volume capacity of flow. In general the D.O.D. has elected to apply heat recovery on most ventilation systems with 2,000 CFM or more of outdoor air flow on a continuous basis. There are several generic categories of heat recovery systems. The simplest is the counter flow extended surface air-air heat exchanger with two air streams separated by a solid barrier which

conducts heat between the streams. This type of system requires the exhaust and make-up air to be brought to close proximity, separated only by the wall barrier of a fraction of inch thickness. These types of systems have no intrinsic active energy consumption elements, exchange only sensible heat, i.e. no mass exchange between streams. The systems have no moving parts and isolate incoming and outgoing streams of air. Their drawbacks are that the two streams must be ducted to a common point, and there is usually a pumping power penalty of fan energy to do this and force the air streams through the exchange surfaces which are usually extended area finned tubes. Heat transfer effectiveness is limited to air-air type conductance values so performance efficiency, i.e. energy exchanged to that available for exchange is rather low. To improve energy exchange requires larger more expensive exchangers and more pressure pumping energy expenditure to drive the flows through the system.

Another type of recovery system is the type which incorporates an intermediate fluid which undergoes a phase-change in interacting with the channels of flow of the two air streams. The air streams are separated and closed to mass interchange but the fluid is an intermediary for heat exchange effectiveness. This may be a so called "heat pipe" with natural convection boiling, and condensation taking place or it may be enhanced by mechanical agitation or pumping to increase heat exchange. The basic improvement is that the two air

flows need not be as intimately connected as in the prior type and the over-all heat exchange can be improved at the same parasitic pumping power input to the air. The latter occurs because the extended air-air friction area is reduced.

A third type of heat recovery system is a variation of the preceeding which splits the incoming and outgoing air stream heat exchanger surfaces to allow stream separation. This is referred to as a run around system with the liquid or two phase fluid loop joining the two separated stream ducts. In general, the circulation of the fluid will require pumping energy as well as the air pumping power input. The key advantage of the run-around system is the allowance for separation of the inlet and exit flows to better meet building layout constraints. Effectiveness of exchange for a given pressure pumping input is about the same or less than for the heat pipe systems.

Another type of heat recovery system is based on an open cycle where the regenerative heat exchanger principle is applied. Typically designed as a circular cylinder, the porous cylinder rotates axially with coaxial air streams passing through separately, but cyclically, through a given sector of the cylinder. As the exit air passes it exchanges its heat energy with the mass of the exchanger and in turn, the mass rotates into the inlet stream to reverse the exchanger. Although the streams do not literally mix, they cyclically occupy the same space and a small carryover of mass trapped in the chamber does

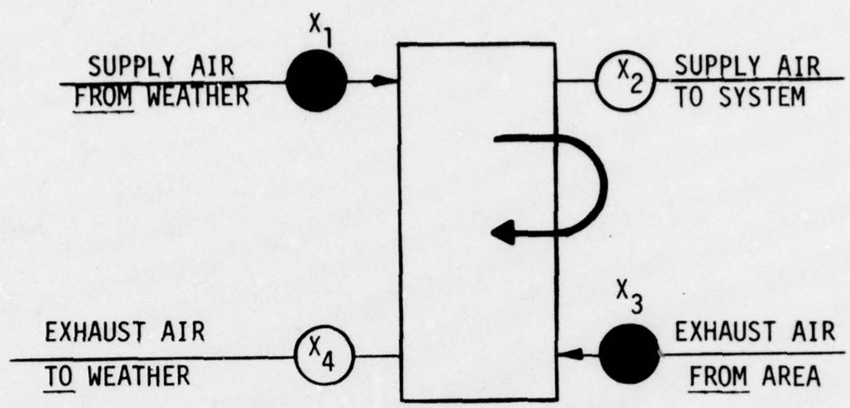
interchange. Therefore some contaminant cross-over possibility does exist. If the porous exchanger is treated with an adsorption material, then mass interchange can occur if concentration gradients exist between inlet flow and exit flow, such as for water content. This type of open system is conducive to sensible and latent energy exchange and therefore is potentially the most effective, provided the cross-over contaminant potential is not deleterious. Some proprietary designs provide for a brief purge of the chambers with outdoor air before allowing the incoming flow to enter the building supply duct. This type of recovery system requires a small amount of power input to rotate the mass, which is usually at a fractional rpm. The air streams require pumping power to pass through the hteat exchanger. The rotating speed, the mass, and the porosity are all design variables affecting effectiveness of operation and design.

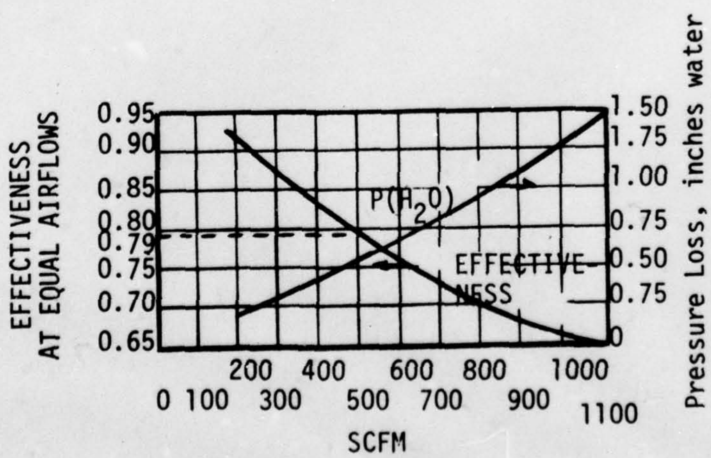
The energy balance analysis is essentially the same for all of the systems with differences arising only in the input of effectiveness coefficient particular to the design of recovery system used. Consider a schematic heat exchange as noted in the following page. Given flow rates Q_s for supply in SCFM and Q_e for exhaust in SCFM, the effectiveness quotient is given by:

$$E = \frac{Q_s}{Q_{min}} \frac{(X_1 - X_2)}{(X_1 - X_3)}$$

where Q_{min} is the lesser of the two, Q's either supply or exhaust. In general, Q_{min} will be equal or less than Q_s so that the Q ratio is greater than unity due to the excess vent flow losses noted in prior discussions. The X's noted above represent temperatures for the sensible heat portion of the exchanger and represent specific humidity for that portion of the heat exchanger which has to do with latent heat effects. The closed system exchangers only display a sensible heat effectiveness. The open system exchangers display both a sensible heat effectiveness and a latent heat effectiveness which may be different in value for a given design with options as to absorber material or porosity. The effectiveness ratio is dependent upon the specific design concept used, and may be as low as 0.4 or as high as 0.85 on temperature i.e. sensible systems and up to 0.8 for latent systems. Some manufacturers display performance effectiveness as a ratio of enthalpy change rather than temperature and/or specific humidity and therefore the proper parameter is needed.

The selection of a heat recovery system also requires a trade-off analysis related to pumping pressure loss of the fans to pass the air streams and that of the energy saved via recovery. A typical heat recovery system will display a recovery effectiveness which is an increasing function of the pressure drop at constant flow. This is because the air side heat conductance is dependent upon high shear levels in turbulent flow, usually over extended surface area (fin-tube)





Typical Operating Performance of Regenerative Open Cycle Heat Recovery System

to achieve good heat transfer. High pressure drop at constant volume flow rate directly affects fan power. Data for one manufacturer to illustrate the cross-over of fan pressure increase and effectiveness with system face velocity is shown in the following curve. Face velocity is essentially the CFM flow per transverse area in ft^2. These and other typical systems operate at about 0.6 inches of water pressure drop on the exhaust and supply sides of the air streams not including any appreciable duct loss but just due to heat exchanger losses. Fan power is then given by:

$$\text{Fan kw} = \frac{\left(CFM_{supply} + CFM_{exhaust}\right)}{6356\ (\eta)} \quad 0.6 \times 0.8\frac{kw}{h.p.}$$

The fan energy used per year can be obtained by multiplying the above average kw power by the fan operating hours per year. This is an energy penalty which reduces the gain from the recovery system. If an intermediate fluid is used between the supply and exhaust, the pump power should be included as well as the additional duct pressure loss.

Using the summer and winter conditions used in the ventilation heat loss example preceeding and a system effectiveness ratio of 0.7 in both sensible and latent heat results in the following energy computation for a ratio of exhaust airflow to supply airflow of 1.20:

$$0.7 = 1.20 \left[\frac{\Delta_T}{44 - 68}\right] \text{Winter}$$

$$0.7 = 1.20 \left[\frac{\Delta_H}{6.4}\right] \text{Summer}$$

The winter heating ΔT from the recovery system is 14 versus the 24 degrees Fahrenheit with no recovery. The summer cooling enthalpy exchange from the recovery system is 3.73 BTU/lb versus 6.4 BTU/lb. The savings in energy is about 58% of the anticipated losses due to ventilation for this case, less any fan/pump power inputs. The savings is therefore:

Energy Saved

$$\text{Winter per CFM} = 14 \times 1.08 = 15.12 \frac{B}{hr}$$

$$\text{Summer per CFM} = 4.5 \times 3.73 = 16.8 \frac{B}{hr}$$

The fan power for a typical fan would be:

$$\text{Fan KW/CFM} = \frac{0.6 \left(1 + 1.20 \right) 0.8}{6356 \times .70}$$

assuming 70% efficiency

$$= 2.37 \times 10^{-4}$$

$$= .8 \frac{BTU}{hr}$$

Therefore the winter savings of 15.12 BTU/hr and the summer savings of 16.8 BTU/hr must be reduced respectively by the 0.8 BTU/hr fan energy per CFM of supply air flow. The energy trade-off is good at a ratio of savings to added usage of about 20:1. However, the cost of the energy saved may not be in the same ratio since fan power is typically electric power and the heat/cool energy cost may be different. For example in cooling at $0.02 per kwhr for electricity and an air cooled condenser system at $2.00 per 10^6 BTU cooling, the cooling savings in

cost is $3.36 X 10^{-5} per CFM and the fan power cost is $4.74 X 10^{-6} for a ratio of cost avoidance to cost incurred of 7.09:1 vs. the energy ratio of 20:1. Therefore the relative cost of energy must be considered for each project in justifying the life cycle cost effectiveness.

A key feature of the economic analysis and justification of heat recovery systems in new and rehabilitation construction is that the installed capacity of the heating and cooling system may be reduced because the heat/cool load at the design point is reduced by the heat recovery system. The influence on capacity will depend upon the use of the building and the locality of the application. If there is a large ventilation heat/cool load proportion in the total building heat/cool capacity then the peak capacity reduction will be substantial. If ventilation heat load is 20% of the total load for a building and 30% of the cooling load then a 58% reduction in vent load by using recovery equipment will reduce installed capacity of heating by 11.6% and cooling by 17.4%. This reduction in capacity to meet reduced peak demands can be substantial in avoided capital expenditure. At a cost of about $2,000 per installation of heating/cooling system, this can be a sizable saving in capital cost as well as an energy savings. Of course the cost of the recovery system must be accommodated. A typical installation would be about $1.00 per CFM of recovery flow for low flows (CFM less than 5,000) and about $0.70 per CFM above 5,000 CFM. Without this cost of equipment capacity savings, the recovery system economics are not as attractive although the energy savings as such are still available. Also, in some areas the use of a sensible heat recovery system only will not be as cost effective as an enthalpy

recovery system because the outdoor conditions are dominated by high summer humidity reduction to achieve indoor comfort levels. This is perhaps best displayed in an example as follows.

Consider application in three different geographical locations for a 20,000 CFM fresh outdoor air requirement. New Orleans, La. is selected as a typical warm humid climate; Bangor, Me. as a cool climate; and Harrisburg, Pa. as a moderate climate but with seasonal extremes. Using average annual hourly climate observations and full time ventilation, a comparison was made of enthalpy recovery systems with the sensible heat recovery systems with the baseline being no heat recovery system at all. Effectiveness of 75% was assumed for each system. The resultant was the following:

	New Orleans		Bangor		Harrisburg	
	Enthalpy	Sensible	Enthalpy	Sensible	Enthalpy	Sensible
Reduced Installed Capacity re: no recovery						
Cooling (tons)	93.4	25.7	42.8	23.0	69.8	27.0
Heating (10^6 BTU/hr)	1.07	0.55	1.76	0.89	1.76	0.89
Annual Operating Energy Savings						
Cooling (10^3 ton hours)	219.0	41.4	3.75	5.4	27.0	16.9
Heating (10^6 BTU)	720.0	346.0	6390.0	2678.0	4554.0	1771.0
System Operating Annual Energy (kwhr)	39200.0	52300.00	39200.0	52300.0	39200.0	52300.0

The above analysis used the following design point conditions as well as the climatic occurence data.

	Summer			Winter		
Location	DBT	R.H.	ΔH(B/lb)	DBT	R.H.	ΔH(B/lb)
New Orleans	94	58	44.6	31	100	11.3
Harrisburg	95	45	40.4	-7	100	-1.1
Bangor	92	38	35.6	-22	100	-5.0

The result to be obtained from an evaluation of this data on operational performance is that the enthalpy type of recovery system is superior to the sensible system when no other type of economizing controls are used. The enthalpy system type of recovery system is an open system and allows some cross-over of exhaust air to the supply. For the above three example applications, the enthalpy system surpasses the sensible system in all criteria. The amount depends upon the location of the facility. In both cases, significant reduction in installed capacity is affected and annual energy savings are appreciable compared with no recovery system. If a system operates less than full time, then the annual savings will be less in approximately the same proportion as the off-time to full-time. However, the installed capacity reduction is still achieved. Installed capacity is a function of the design point condition of outdoor conditions and building demand. Therefore the equipment must be sized to this even though the condition occurs only infrequently. Therefore an economic advantage exists to minimize the installed capacity of equipment. This results in reduced chiller capacity, and fan

capacity. Both are usually electric driven and therefore will accrue peak demand utility charges in proportion to size. This means that even if the recovery system only operated on the peak demands days that cost savings in initial and operating costs will be achieved independent of sustained energy (cost) savings through continuous use of the equipment. The functional way of evaluating this is to examine the coefficient of performance COP as a chiller replacement. A typical mechanical chiller will give a COP of 4 or less. This is the ratio of cooling achieved per energy input. The heat recovery systems have a COP equivalent of 50 or greater. Therefore the thermodynamic effectiveness is outstanding compared to adding chiller capacity. For heating, there is no equivalent to COP, however in the prior example calculation, it was shown that the energy saved by a typical recovery system is about 20 times the necessary input energy to operate the recovery system. It is very difficult to find active systems that can match or exceed this type of performance. Considering only the cost of the chiller at about $350 per installed ton of cooling, the recovery system will save, for typical example shown previously for Chicago as follows:

$$\text{Installed Cost} = \text{Saving on Chiller Reduction} \quad \frac{\$350 \text{ X } 16.8 \frac{\text{BTUH}}{\text{CFM}}}{12\text{MBTUH}} = \$0.49/\text{CFM}$$

$$\text{Installed Cost} = \text{Saving on Furnace/Boiler Reduction} \quad \frac{\$75 \text{ X } 15.1 \frac{\text{BTUH}}{\text{CFM}}}{33000\text{MBTUH}} = \$.03/\text{CFM}$$

Therefore about \$.52 per CFM in deferred initial cost offsets from ½ to 2/3 of the initial cost of the heat recovery system. This does not include any effect on demand charge reduction, energy consumption savings, or other effects. The economics of each installation will be location specific and building use specific since the humidity and design DBT's dictate the potential savings in installed capacity and consumption.

An energy recovery system used on a building with partial occupancy per week and vent shut-down during unoccupied periods will experience less consumption based savings than a full-time use building. Also, the alternative of using an economizer cycle to use "free outside air cooling" will also reduce the potential for consumption savings. Each application should be carefully evaluated as to whether a recovery system will provide significant consumption cost savings over the useful life of the system. In general the heat recovery system will pay for itself in deferred energy cost at some future time. The only concern is to evaluate whether the investment in capital warrants the savings in fuel compared to other options at this time. Heat recovery systems can be expected to be extremely viable for continuous, or nearly so, use buildings with high ventilation rates and air changes. This would include: most recreational buildings, hospitals and clinics, food service areas and barracks. These facilities have high CFM requirements for outdoor air and are continuously occupied. Locations with high latent heat variation between the outdoor air and

the interior space are particularly good candidates for the enthalpy type of exchangers. These include swimming pools and controlled environment rooms for electronics in humid climates where latent loads are large in cooling.

It is not a simple generalization to estimate the annual energy savings of a system for heat recovery since occupancy and location are important as noted in the previous examples. With increasing fuel costs, the ultimate result is that recovery systems will be viable options for all but the non-ventilation control buildings at some future date. Therefore almost all new and rehabilitated construction should incorporate the heat recovery systems if at all feasible and take advantage of the reduced capacity noted in initial equipment sizing. The consumption savings will pay for any residual over a period of time depending upon occupancy and the full cost of the saved energy.

The types of heat recovery systems available for application are of the following categories:

a) rotary, recuperating, heat exchangers referred to as heat wheels, with or without dessicant capability for enthalpy exchange. These are open cycle systems.

b) air-air solid metallic heat exchangers, typically with extended area surfaces and often in cross-flow. These do not mix the two streams and are primarily of the sensible heat exchanger type.

c) heat pipe systems with air-fluid-air heat exchange and a phase change to enhance heat transfer and to ease the problems of making the flows coincide for heat transfer contact. These may also incorporate active agitation of the fluid.

d) split systems with the exhaust air and make-up air sources separated in distance and an intermediate fluid with or without phase change in the pumped fluid. These are cloded to stream mixing and are often referred to as run-around systems.

e) mechanical enhancement split systems, which include a heat pump principle to achieve a lift in temperature difference between the two flows and further enhance heat transfer. In this system, a refrigerant type of fluid is used and the air streams are the source and sink for heat exchange. The operating energy of these systems are higher than any of the others.

Each of the above type of systems have their own place in the application spectrum. The first criteria is probably that of the capacity requirement. In small vent flows of less than 5,000 CFM the initial cost and operating cost may tend to the simpler less effective heat recovery system such as a type b or c above. At larger capacity and continuous duty the others will become more attractive. The type a is applicable to those situations where mixing a carryover of some small portion of the

air or contaminants is not a large concern and particularly if enthalpy recovery is warranted i.e. high latent heat differences indoor to outdoor. It is now possible to obtain average annual DBT and WBT occurrences for CONUS cities and therefore to obtain the outdoor air enthalpy occurrence on an hours per year basis. This data allows calculation of the relative effect of attempting to recover both sensible and latent heat or sensible only for a given location and application. The choice of an integrated or a split system (run-around) will depend upon the ease of achieving close coupling of the prevalent exhaust vents and intake grill. In general, the cost and energy effective solution will be to use the liquid interconnect loop rather than duct long runs of air flow to a common point. The active system, i.e. the heat pump is only practical on very large flow applications with relatively small enthalpy differences indoor/outdoor and steady operation. In effect, and additional machinery cost of acquisition and operation is being applied to achieve higher heat exchange effectiveness. The type e system should be carefully analyzed before commitment.

VENTILATION ENERGY LOSS CALCULATION

Example Calculation--Split System Heat Recovery System

The calculation of the energy balance for a heat recovery system is much the same as for any air-air heat exchanger. The split system or run-around system where the supply (outdoor air) and the exhaust air interchange first with an intermediate fluid is slightly different and worth explaining in depth. The systems using a phase changer, heat pipe, principle without pumping are subject to the specific design and thermo-dynamic properties of the fluid. The concept and the fluid may be priopprietary to the manufacturer and so details are not possible to generalize.

The system selected for example uses an ethylene-glycol pumped fluid between two finned coils, one in the supply air and one in the exhaust air. A schematic diagram is presented below defining terms. In this example, the outdoor air temperature and the (exhaust) air return from the controlled space are selected. In the example, the duct size of the exhaust and supply air are assumed for coil size limitation. The finned coils are typically inserted directly into the duct flow analogous to a reheat hydronic coil. The following data was assumed for the example:

Mixture 70% water 30% glycol

t_{si} = 20F

t_{ei} = 70F

$SCFM_s$ = 6400 CFM

$SCFM_e$ = 6000 CFM

Supply coil face size (duct) = 24" X 78"

Exhaust coil face size (duct) = 24" X 72"

The ratio of supply air to exhaust air is:

$$M = \frac{6400}{6000} = 1.07$$

Estimate the fluid flow rates needed from the relative heat capacities, i.e. $Q/\Delta T$

$$\frac{Q}{\Delta T_s} = 1.08 \ SCFM_s = 6912 \ \frac{BTU}{HR^oF}$$

$$\frac{Q}{\Delta T_e} = 1.08 \ SCFM_e = 6480 \ \frac{BTU}{HR^oF}$$

The average of these two is 6758 which is a nominal target for the glycol loop. The glycol mixture at an average of 45^{o}F has a specific heat and density product such that

$$\frac{Q}{\Delta T_g} = 464 \ GPM \qquad \text{where GPM is the gylcol flow rate}$$

The averaged needed flow rate is:

$$GPM = \frac{6758}{464} = 14.6$$

Finned coils are typically available in a variety of rows of depth as well as the area of the face. The hydraulic piping arrangement of parallel and series flow is such as to develop good heat transfer at nominal pressure drops. In this size of coil, 2 GPM per circuit is a good average.

This would indicate 7.3 circuits. Actually 8 circuits would be required to round-up. Therefore the glycol flow rate must be approximately 16 GPM.

The face velocity of the coils is as follows:

$$V_s = \frac{6400 \text{ CFM } 144}{(24 \text{ X } 78)} = 492 \text{ FPM}$$

$$V_e = \frac{6000 \text{ CFM } 144}{(24 \text{ X } 72)} = 500 \text{ FPM}$$

These are typical optimal face velocities to achieve good heat transfer at acceptable pressure drop of air flows. Using a curve, attached, the overall conductance (q) of the exchanger system has been presented for a specific model series of finned coils. Knowing the face velocities and relative flows, the q is available from the curve. For this example the q = 355. The heat recovery is then given by:

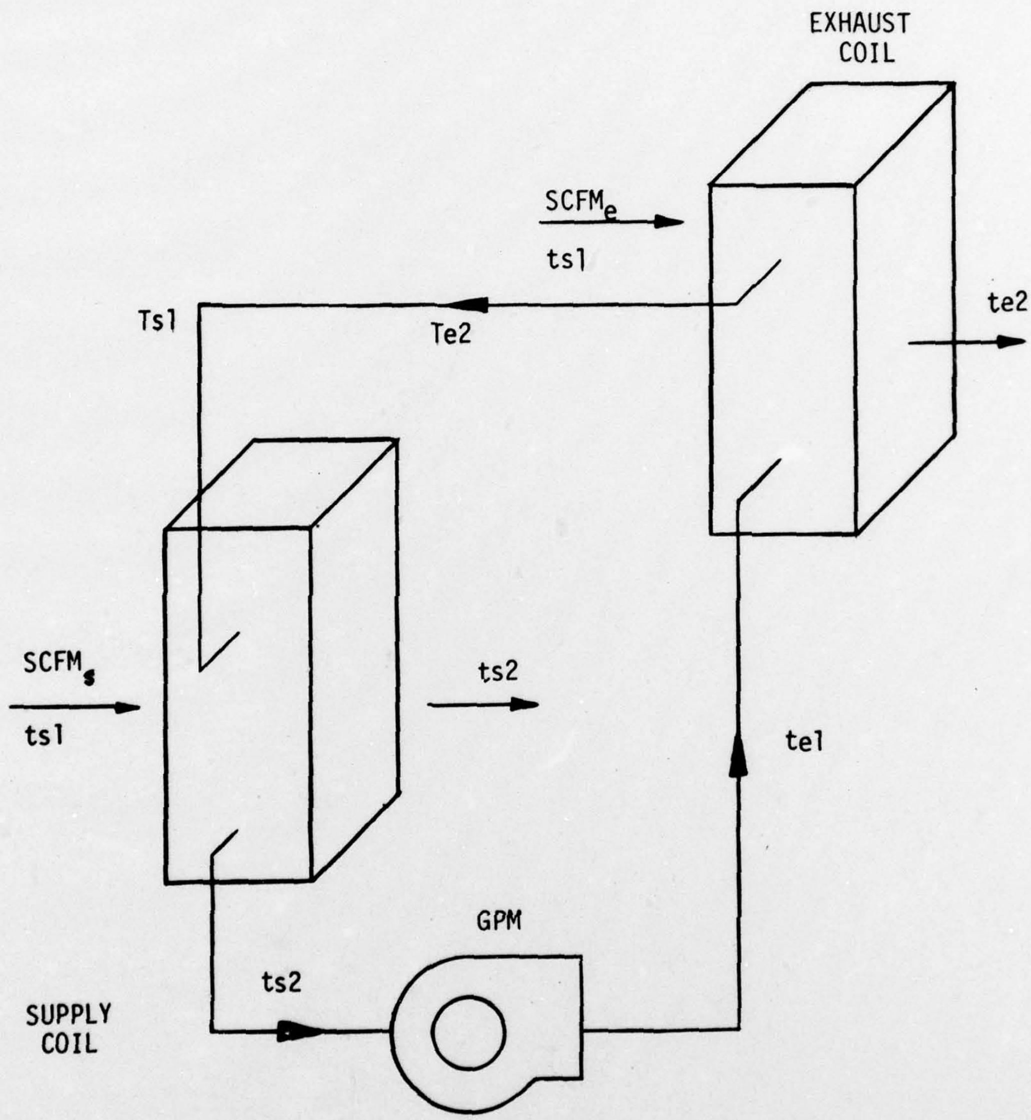
$$Q = q(\Delta T)A_e = 355\ (70-20)\ \frac{(24 \text{ X } 72)}{144} = 213{,}000$$

The leaving exhaust air temperature is given by:

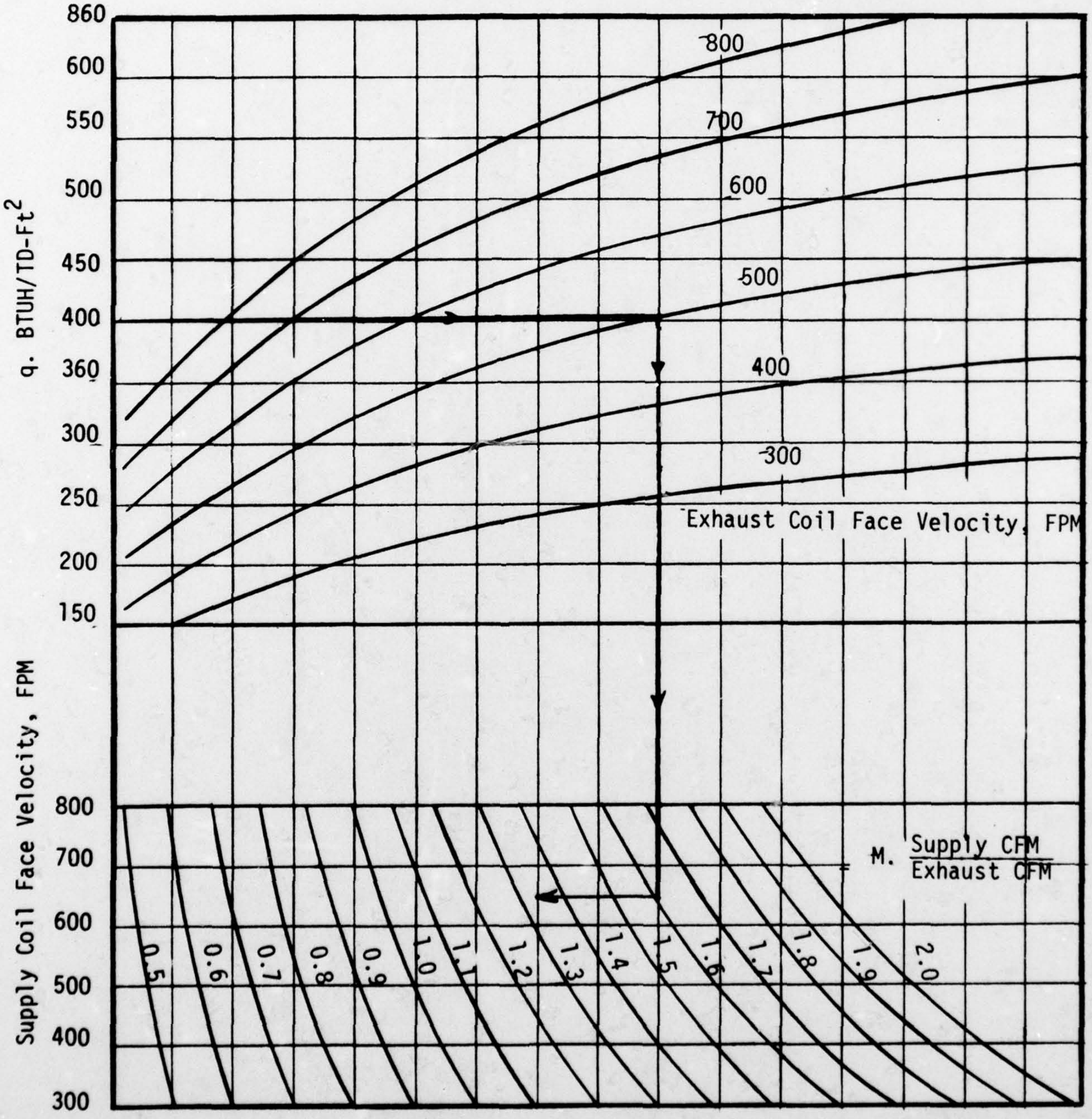
$$T_e = \frac{Q}{6540} = 32.6^{o}F,\ T_e = 70-32.6 = 37.4^{o}F$$

$$\Delta T_s = \frac{Q}{6976} = 30.5F$$

$$T_s = 20 + 30.5 = 50.5F$$



Typical Run-Around Heat Recovery System



Typical Run-Around System Operating Performance.

The above assumes no loss of heat in the piping system and the coil characteristic selected i.e. the 5 row coil. Also, no latent energy was assumed to be exchanged. The pressure drop of the water side and air side are also important design features which dictate pumping power. For this example the flow drops were computed to be:

$$\Delta P_s = 1.3 \text{ in } H_2O$$

$$\Delta P_e = 1.35 \text{ in } H_2O$$

$$\text{glycol } \Delta P = 16.8 \text{ ft } H_2O \text{ (exchanger only)}$$

The glycol temperatures were found to be:

$$T_{s1} = 58.6F$$

$$T_{s2} = 29.9F$$

The overall effectiveness as a measure of the ΔT actual to the supply-exhaust temperature achieved is:

$$\frac{\Delta T}{70-20} = \frac{50.5-20}{70-20} = .61$$

Therefore the inlet air is raised to 61% of the exhaust air potential. Note that the flows are not equal and that this system would perform better if they were. The summer performance on sensible heat only would be, for $95^{o}F$ outdoor and $78^{o}F$ indoor, approximately 72.4 MBTUH. For this example, note the power input requirement to operate the system. Assume a 0.70 efficiency for all fans, pumps and neglect duct

and pipe loss other than the coil drops.

E (supply fan) = 1.49 KW

E (exhaust fan) = 1.46 KW

E (glycol pump) = $\underline{\quad .5}$ KW

$\sim$ 3KW

This amounts to an input of about 10 MBTUH per hour of operation and a yield of up to 213 MBTUH for a COP of about 21. However, this is at winter design conditions and often the outdoor temperature will be higher and provide less of a yield. To conduct an economic analysis requires the enthalpy (or even DBT) occurrence in hours per year to determine the practicality of the system, particularly if part time use is anticipated due to occupancy.

R&D REQUIREMENTS DEFINITION

R&D Requirements Definition for Future HVAC Equipment and Systems for Energy Savings

In the future years, energy costs will escalate more rapidly than the economy at large. This has been described in an earlier section of this report under Cost Models. Therefore observations as to energy conserving improvements in HVAC equipment and systems can consider items which in the recent past have been overlooked as not being cost-effective. Another aspect to be considered in R&D goals is that the Government buildings average age is increasing and many will require either major rehabilitation or replacement. Construction costs and congressional allocations tend to be such that rehabilitation will be the most prevalent method of maintaining the total facility space in economical operational condition. The goals for R&D in HVAC for rehab needs will definitely be different than those for new construction. For example, new construction will probably be integrated to have optimal insulation, minimal ventilation (sealed design) and architectural compatibility with the use of active or passive solar energy modes perhaps in conjunction with heat pumps and/or absorbtion cooling equipment. This study did not address either solar energy or heat pumps in depth primarily because of the specificity of the concept to the building siting and mission. The R&D goals presented are primarily those associated with projects that could be applied in the majority of future rehab and typical construction situations.

The major categories for R&D in HVAC equipment ans systems for cost-effective energy savings break down into: control strategy develop-

ment; ventilation energy savings via lower throughflows and recovery of outbound energy; specific equipment improvements such as fan volume control, terminal boxes, etc. and also design to energy effectiveness through the use of existing programs for optimal pipe insulation, duct sizing, pumping requirements, etc. Within each of these is a whole spectrum of practicality. For example, mechanical (vapor compression) chiller coefficient of performance (C.O.P.) can be improved through various developments in compressor design, use of screw and centrifugal compressors of high efficiency, etc. However, the cost of such R&D to effect the improvements will be of the $100 million category. If this is not borne by the manufacturers of the equipment, then the Government will participate through direct aid or tax subsidy in the selling price re: efficiency discounts. The time to reflect an improvement in the Army building energy status would be many years after the R&D, because the existing equipment in many facilities, although relatively inefficient by latest standards, is still durable and functional for many years. Natural attrition of these units would require perhaps 20-30 years to put new R&D improved equipment on-line in 95% of the applicable projects. The first priority of R&D will best be directed to generally applicable low capital cost elements that can rapidly be integrated into the inventory of Army buildings to effect energy savings.

Timed use controls are generally available already and are in use in many buildings. These apply set-backs or shut-off of non-essential HVAC equipment during unoccupied periods. The next step in R&D to timed use controls is to integrate with small scale or large scale energy

management and control (EMC) computer and software equipment. The latter are also available and practical for large facilities. Single building and small facility sites are not currently conducive to cost effective application of EMC equipment of the complexity that is prevalent. In renovation and retrofit of facility HVAC equipment for timed use and sequence, one problem is usually that of the cost of providing control hardwire to the equipment sites from a central sequence. At the present time there are alternates available to hardwire. One uses pulse code modulation along the power supply to trigger remote receivers into action or inaction. This approach is generally good except that it is subject to false action due to natural transients in the line resulting from other switch closures, etc. Another method is to use an FM radio signal to remote switches. This system is subject to saturation when many buildings nearby are using such transmissions. R&D to arrive at reliable, inexpensive master-slave communications to install in existing buildings with minimal wiring or parasitic energy drain is a desirable goal. The specific sequence of operation can be as simple or complex as the application requires. For example, one manufacturer already markets a simple logic sequencer for small administrative and training type buildings which senses outdoor air temperature and wind, the indoor air and clock-time to make a decision as to when to initiate early A.M. heat/cool and vent functions to just arrive at a minimal comfort at occupancy. This system, coupled with remote actuators assures that night set-back recovery is only initiated to meet the real-time need and not a preprogrammed time

(e.g. 4:30 A.M.) selected to meet a maximum design condition. This logic is also programmed to provide extra heat-up time on Monday A.M. to accomodate weekend soak-down. A typical simple logic and sensor system will cost about $1,000 but hardwire to the equipment is extra. An integrated package suitable for multi-zone application in small to medium sized office buildings or other periodically occupied space would be highly desirable for optimal set-back control without recourse to an EMC package of greater sophistication. Simple pay-back times of the order of 1½ to 2 years are projected and many existing building systems could be so equipped in a very short time, i.e. 2-5 years to rapidly implement the savings. This approach is essentially a second generation unoccupied temperature-vent air set-back concept. The R&D required is two-fold. The application statistics as to how many zones and operations may be controlled with the simple logic relative to the EMC approach in what proportion of Army buildings to arrive at a common package to have greatest effect. Next, the signal transmission-reception system used to relay commands must be proven to be reliable, inexpensive, and simple to install.

Thermostatic operational controls are another area warranting R&D implementation. The future rehab and new installations of HVAC equipment will best incorporate so called "economiser" controls on vent air and also many will be of the variable air volume (VAV) type. Therefore thermostatic circuit controls to provide optimal hot and cold deck reset temperature to meet demand, and also the selection of the fan volume control decision processes for pumping energy reduction are significant.

The basic equipment is already available. The R&D requirement is to develop logic systems to operate on the input data to select best conditions without recourse to expensive computational equipment. This requries a reliable simple logic panel development at the next level of sophistication to the now prevalent temperature sensing motorized damper controls. Again, the objective is to provide a common low cost system element capable of rapid assimilation into the application Army building inventory.

Outside of improved control strategy, the next area of R&D which promises improved energy efficiency is that of ventilation reduction and improved energy economy for ventilation. First, ventilation should only be used during the period of use when occupied or when potentially hazardous contaminant build-up is likely due to operating equipment or processes. New ASHRAE standards suggest substantially less fresh air make-up than was past practice. In some cases the reduction is almost 10 fold. First order changes are possible without R&D, namely to use better filtration and rely less on purging to reduce contaminant levels. After these obvious reductions have been made, then improved equipment will be in order. There is a point of diminishing return on improved filter equipment. At some point it is less expensive and less energy consuming to apply heat recovery equipment to recover 60-70% of lost ventilation energy than it is to further reduce fresh air intake and to vastly increase the amount of air clean-up that the filters must achieve to provide a satisfactory supply of clean air. This limit appears to be at the 3-5 CFM per occupant level for most applications. Below this it

is necessary to achieve high filtration efficiency and to provide for odor and/or chemical contaminant removal to a high level of efficiency. The current R&D requirement is to develop odor removing equipment which does not have potentially hazardous side effects, i.e. ozone generation, chemical vapor exposure and other health hazards. The development of surface absorbant agents for impregnation upon or integrated into particulate filter media would be a practical objective for R&D.

In the area of ventilation heat recovery, the problem of retrofit is primarily that of inappropriate building design rather than an R&D need. Many buildings have distributed vent systems for air exhaust. The heat recovery system will be very small in any one vent location or else require a run-around loop to collect the heat at various discharges and input the various intakes of air. A typical efficient control scheme for outside air control is presented in the Figure 4.

Ventilation air clean-up system R&D should review the air-washer approach to filtering air. The ability to wash impurities from the air of both chemical and particulate type is a distinct advantage. The attendant humidification in winter and dehumidification in summer that takes place with the use of appropriate water temperature is an added feature of this type of system. The R&D features available for improvement are the type of absorbant liquid used to clean the air. Water alone can only remove particulate and soluble odorants. Active agents can actually oxidize or otherwise react the contaminants to passive byproducts. This is primarily a chemical engineering process problem coupling economics

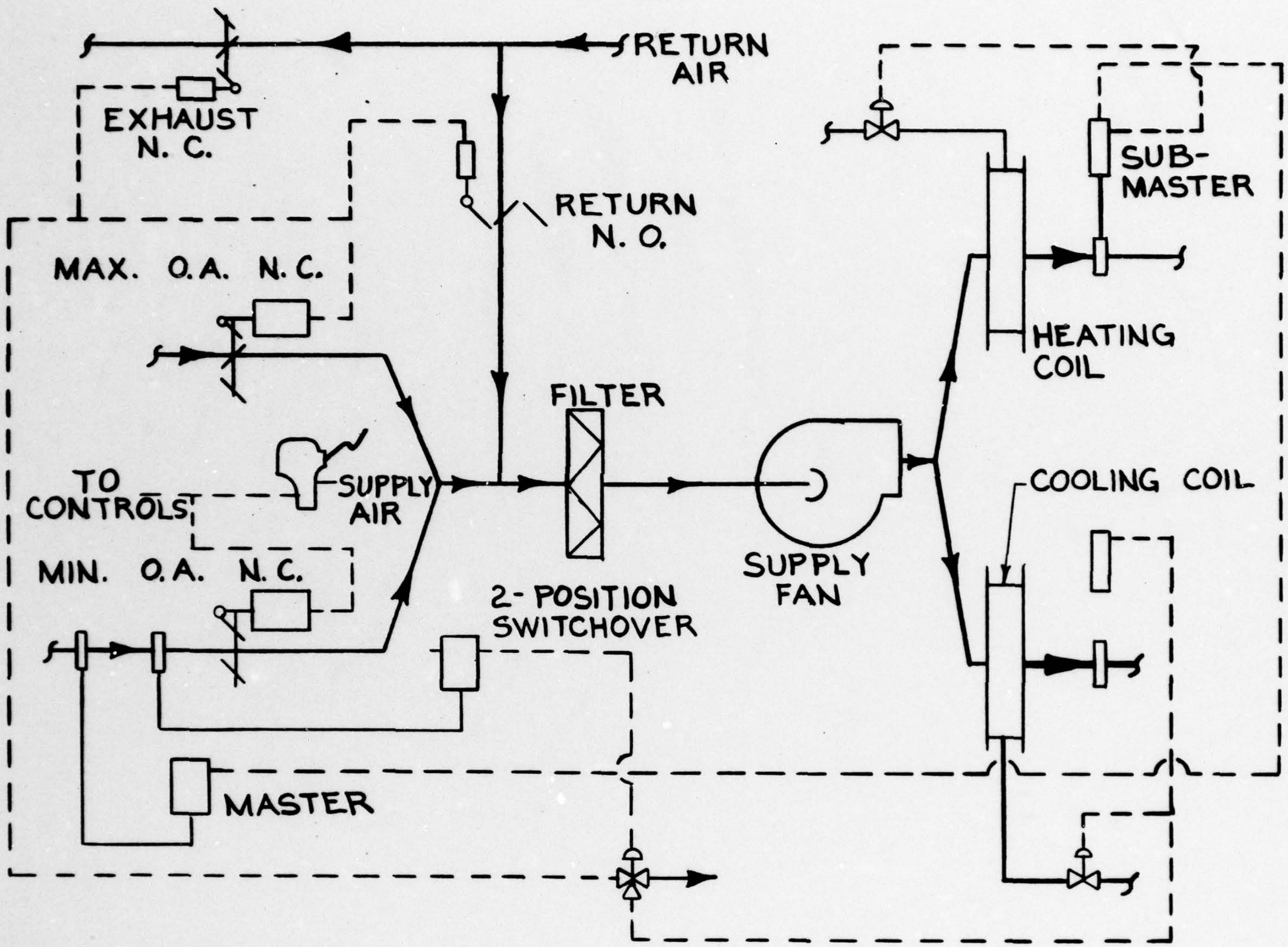


Figure 4. Typical Economizer System Diagram.

with chemical operations. The reactant material should be simply regenerated or easily disposable without environmental problems. The vapor of the active agent should also be safe for continuous human exposure at the levels developed by the equipment operation. The technology of separating water droplets out of the thru flow of air is also in need of improvement. The removal of essentially all liquid water is required to be accomplished with minimal energy input. The impinging barrier type of separator has high relative losses as do centrifugal separators. A high efficiency separator with low energy losses is required.

Equipment R&D goals and objectives should be directed toward the improvements in terminal VAV box operation at variable volume flow. Too many VAV boxes do not provide adequate control to accomplish the amount of energy savings possible with this improved system. The VAV flow control in self (pressure) powered boxes requries R&D to achieve better control at saturation, i.e., low duct pressure. In start-up and recovery from night set-backs the VAV boxes may be calling for wide-open fan flow. The system may have been volume flow sized for 80% of total design load since not all zones would have been anticipated to have simultaneous peak demands. At wide open demand by controls, the duct static pressure will be very low and the self powered boxes flow dampers will not function to control. The fan will be called upon for over design flows, which could overload equipment depending upon fan blading and motor capacity, etc. To achieve the economy of small size and low pumping power usually associated with the VAV system it is necessary to develop flow control boxes which can maintain flow in control independent of duct pressure within usual limits. The

variable position, variable area venturi type box concept to adjust flow rate a near constant total pressure, i.e., low static loss, is an energy efficient approach which will allow the desired control authority to be achieved. These boxes could be incorporated into existing or currently designed retrofit projects in the near future to effect energy savings. The R&D objective is to develop the variable flow box with the necessary self-powered control at low loss of total pressure and at high flow. This will allow the system to control to maximum flow, consistent with fan capacity, without starving distant boxes of flow and without requiring high static pressure duct systems. This R&D outcome would allow early VAV designs to be made compatible with the newer requirement for set-back recovery in the A.M. which often takes the control system out of authority due to low duct pressure.

The variable volume output fan for use in VAV sir systems will become a significant item in the pumping energy budget of the system life cycle cost. The Variable Volume Flow Fan has been discussed in some detail in the preceding sections. There are currently several modes of fan output volume control in use. Variable inlet vane is one of the best compromises between simple damper use and a complex variable rotor blade geometry or motor speed variation or combinations of them. R&D in the field of small fan aerodynamic efficiency improvement for a long term goal could be achieved in a few years so that projected rehabilitation for energy conservation could use them in the 5-10 year future. If conversions are

made now and for the next 5 years in large numbers, the development of better VAV fans could come too late to achieve energy reduction for many years hence until the next cycle of new construction and rehabilitation occurs.